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ABSTRACT

This paper will compare relative part-load efficiency of screw and centrifugal compressors. Compressor part-load performance will be presented in the form of a two-dimensional performance map showing head on the vertical axis as a function of flow on the horizontal axis with efficiency islands indicating efficiency for each possible head/flow combination. This way of presenting compressor performance is common for centrifugal compressors but novel for screw compressors. The compression efficiency comparison will focus on the vapor compression efficiency. Mechanical and/or electrical losses from bearings, transmission, drive and - in the case of variable speed drive – frequency converters have to be added to obtain a valid overall compression efficiency comparison.

1. INTRODUCTION

Compressors are selected for full-load design operation, which is commonly defined as the condition of maximum required capacity and pressure rise. Most of the time, however, compressors will run at operating conditions corresponding to a lower flow rate and a smaller pressure ratio than the original full-load design point. These off-design conditions always result in reduced compressor power consumption which might hide the fact that compressor efficiency typically deteriorates at off-design conditions.

Water-cooled chillers in the smaller capacity range of 0.5 to 2.0 MWth of refrigeration, which traditionally used centrifugal compressors, have during the past two decades seen a larger utilization of screw compressors. Claims of better part-load efficiency and the absence of surge have promoted the introduction of screw compressors as an alternative for centrifugal compressors for these applications. The use of variable speed operation replacing the slide valve or puppet valve control methods has further improved screw compressor efficiency [1]. On the other hand, the recent commercial introduction of small high-speed, direct-drive centrifugal compressors has targeted the screw compressors in the 200 – 500 kWth capacity range, again claiming better part-load efficiency as the motivation to switch from screw to centrifugal compression technology [2].

A previous study analyzed the relative benefit of variable speed operation for various centrifugal compressor concepts used on a water-cooled chiller [3] by comparing the location of peak efficiency islands on the compressor performance map to the ARI load line.

The purpose of this paper is to compare relative part-load efficiency of screw and centrifugal compressors both in fixed-speed and variable-speed configurations and to determine which load lines are best served by which type of compressor. Compressor part-load performance will be presented in the form of a two-dimensional performance map showing head on the vertical axis as a function of flow on the horizontal axis with efficiency islands indicating efficiency for each possible head/flow combination. This way of presenting compressor performance is quite common for centrifugal compressors but novel for screw compressors. The compression efficiency comparison will focus on the vapor compression efficiency. Differences in off-design compressor efficiency show up as differences in the location and shapes of the efficiency islands on the compressor performance map.

Screw and centrifugal compressors have completely different loss mechanisms.

The major loss mechanisms of the screw compressor are: leakage loss, oil drag loss, exit port flow loss and oil separator pressure drop loss. The first two loss mechanisms are independent of capacity and will become relatively more dominant at lower flow rates. The last two losses are proportional to the square of velocity and become less pronounced at lower flow conditions. The built-in volume ratio of the screw compressor results in over- or under-compression losses at lower and higher pressure ratio than design, respectively.

The main loss mechanisms of the centrifugal compressor are friction and flow diffusion losses. Incidence, choke and additional diffusion occur at part-load conditions. Impeller disk friction is the constant parasitic loss for a constant speed centrifugal chiller that becomes more pronounced at part-load conditions. Flow friction losses will reduce at lower flow rates.

Based on these very different loss mechanisms the shape and location of the efficiency islands on the compressor performance map is expected to be very dissimilar for screw and centrifugal compressors.
The generic shape of the performance map of five different compressors will be shown and compared in this paper:
1. Fixed speed vaned-diffuser centrifugal compressor with inlet guide vanes
2. Fixed speed vaned-diffuser centrifugal compressor with variable diffuser geometry
3. Fixed-speed oil-flooded screw compressor with slide valve
4. Variable speed low-oil screw compressor .
5. Variable speed vaned-diffuser centrifugal compressor with variable inlet guide vanes

Mechanical and/or electrical losses from bearings, transmission, drive and - in the case of a variable speed drive – frequency converters have to be added to obtain a valid overall compression efficiency comparison.

2. CENTRIFUGAL COMPRESSOR PERFORMANCE MAPPING

A centrifugal compressor running at constant speed can produces a range of pressure ratios at varying flow rates. Maximum compressor flow is achieved at low pressure ratio. At higher pressure ratios the flow rate starts to diminish. Eventually, a minimum flow/maximum pressure ratio combination is reached beyond which the compressor will surge. For each point on this curve we can define the compressor efficiency as the ratio of the required compressor enthalpy rise if the compressor process were loss-free and the actual enthalpy rise:

$$\eta_{is} = \frac{\Delta h_{is}}{\Delta h_{act}} \tag{1}$$

For given compressor inlet temperature and pressure and a specified exit pressure, the loss-free ideal compressor enthalpy rise Δh_{is} can be derived from thermodynamics by assuming an isentropic compression process:

$$\Delta h_{is} = h_2(P_2, s_1) - h_1(P_1, T_1) \tag{2}$$

Actual compression enthalpy rise can be derived from

$$\Delta h_{act} = h_2(P_2, T_2) - h_1(P_1, T_1) \tag{3}$$

Note that centrifugal compressor efficiency can be determined solely from temperature and pressure measurements without any power or flow measurements.

Centrifugal compressor performance is often presented in terms of head as a function of flow rate. Isentropic compressor head H_{is} is defined as:

$$H_{is} = \int_1^2 V(s_1, P)dP \tag{4}$$

The path of compression is uniquely defined as the isentrope starting from a given initial temperature and pressure to a final pressure. Isentropic head equals isentropic enthalpy rise.
The actual rise in enthalpy is equal to the input head H_{in} of the compressor. Compressor input head can therefore be defined as:

$$H_{in} = \frac{H_{is}}{\eta_{is}} \tag{5}$$

Centrifugal compressor input head can also be derived from the Euler turbine equation as the change in angular momentum of the flow:

$$H_{in} = u_2 c_{2,\theta} - u_1 c_{1,\theta} \tag{6}$$

Figure 1 shows the relationship between isentropic head or enthalpy rise and flow rate for a centrifugal compressor with a peak efficiency of 85%. The compressor efficiency varies as function of isentropic head and flow rate. Its peak value occurs near the "knee" of the curve. Compressor input head or actual enthalpy rise can be obtained by applying Equation (3) or (5) by dividing these isentropic head or enthalpy rise values by the corresponding efficiency values. The Euler input head equation (6) explains the straight-line characteristic of the input head curve at fixed-speed operating conditions (constant impeller tip speeds u). From Figure 1 the compressor loss at a given flow can be interpreted as the distance between the input head and the isentropic head relative to the value of the input head for that flow rate. Peak efficiency is realized when the isentropic output head approaches the input head, i.e. at high pressure ratio, and drops off almost linearly with a reduction in output head since the input head does not reduce when the output head goes down. It becomes also clear why a centrifugal compressor is limited in terms of its maximum deliverable head or pressure ratio since the input head is fixed.

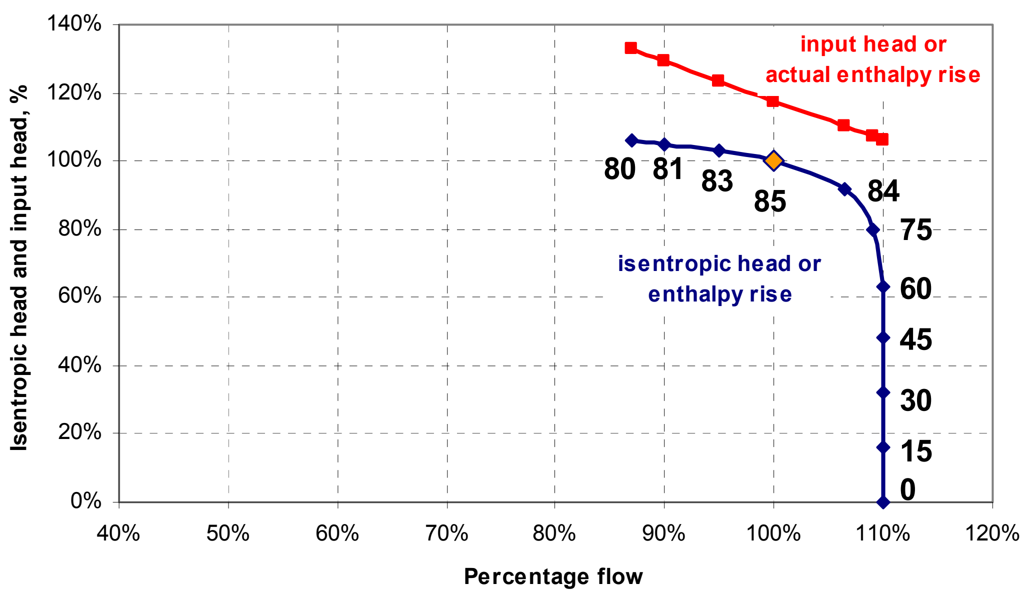

Figure 1. Head flow characteristic of a centrifugal compressor

Continuous off-design operation of centrifugal compressors can be achieved by the use of variable geometry compressor hardware. Inlet guide vanes that can adjust the amount of inlet pre-swirl the impeller sees and variable geometry diffusers that can modify the diffuser passage width or throat area are devices that are used for continuous compressor operation under varying conditions. The single one-dimensional head/flow characteristic shown in Figure 1 is replaced a two-dimensional area of possible compressor operation that can be achieved with e.g. different inlet guide vane setting angles. By connecting points of equal compressor efficiency for the different inlet guide vane setting angles a performance map with efficiency islands can be obtained. Figure 2 shows a compressor map obtained in that fashion. The boundaries of the map indicate the largest pressure ratio/head that can be achieved at a given flow rate before surge occurs (surge limit) and the largest capacity that can be obtained at a given head (choke limit). The dashed lines in this figure are the normalized efficiency islands. They show the relative change in efficiency at off-design conditions.

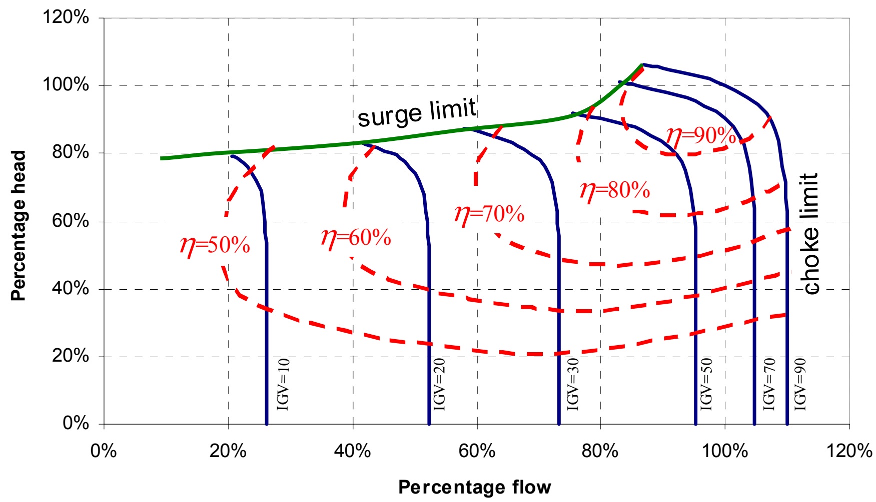

Figure 2. Performance map of a centrifugal compressor with variable inlet guide vanes

3. SCREW COMPRESSOR PERFORMANCE MAPPING

Screw compressor efficiency can not be obtained experimentally from pressure and temperature measurements alone. The oil, used for lubrication and sealing of these machines, affects the compressed vapor exit temperature. This makes it impossible to determine the exit enthalpy from pressure and temperature measurements.

Instead, positive-displacement compressor efficiency is determined as the ratio of ideal compressor input power to actual input power:

$$\eta_{is} = \frac{\dot{W}_{is}}{\dot{W}_{act}} \tag{4}$$

Actual input power is obtained either through torque and speed measurements or – as is more typical - through the measurement of input power of calibrated motors. The motor input power measurement includes the motor inefficiency of the compressor. Actual compressor work $\dot{W}_{act}$ be obtained from measured motor input power after adjusting for motor efficiency:

$$\dot{W}_{act} = \eta_{motor} \dot{P}_{motor} \tag{5}$$

Measurement of the compressor mass flow rate $\dot{m}$ is required to determine isentropic compression power from isentropic enthalpy rise:

$$\dot{W}_{is} = \dot{m} \Delta h_{is} \tag{6}$$

As a result of the different method of determining compressor efficiency, mechanical loss is lumped together with the gas compression efficiency in positive-displacement compressor efficiency ratings while it is often separately accounted for in centrifugal compressor rating procedures.

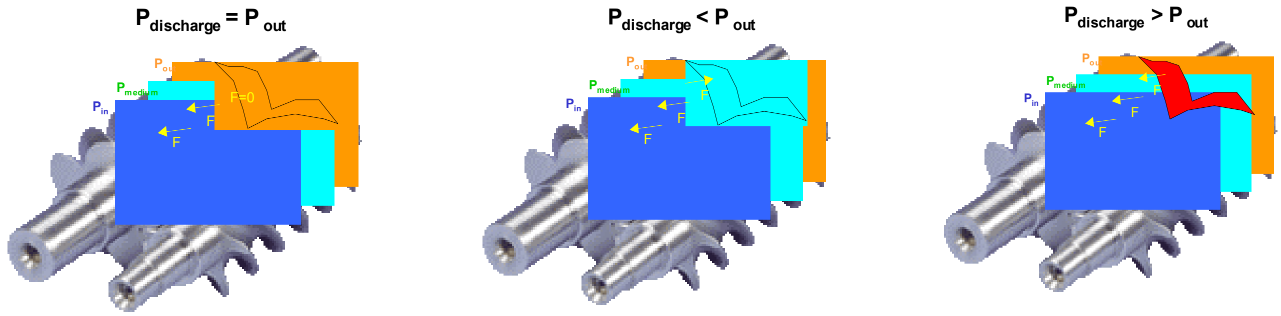

Figure 3. Effect of screw compressor discharge pressure on work input power

The screw compressor work input for a given flow rate varies with head. This is contrary to the constant work input for a given flow rate of a centrifugal compressor. The change in compressor work input with head or pressure ratio can be understood from Figure 3 where the pressures of the last three enclosed flutes of a screw compressor are shown with the last flute discharging the compressed vapor at the following three discharge pressures:

a. the design discharge pressure
b. a lower than design discharge pressure
c. a higher than design discharge pressure.

It is seen from Figure 3 that at lower exit pressure than design (during so-called over-compression relative the required pressure) the forces are reversed over the last lobe resulting in some expander action that will reduce the compressor work input. Similarly, at high discharge pressure the last flute will be filled with high pressure vapor the moment the flute reaches this discharge port area (under-compression) causing an additional force over the last lobe of the compressor, thus increasing the compressor work input. The increase in compressor work with higher than design discharge pressures allows the screw compressor to operate stable at higher pressure ratios than design. The reduction in work input at lower pressure ratios improves the low pressure ratio efficiency compared to a fixed speed centrifugal compressor. These two inherent advantages of screw compression technology over fixed speed centrifugal compressors have helped screw compressors make inroads in centrifugal chillers, despite their traditionally lower design efficiencies.

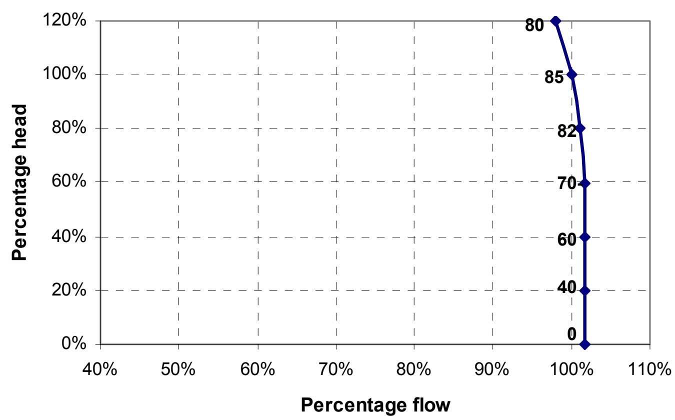

Figure 4. Head flow characteristic of a screw compressor with isentropic efficiencies varying less than linearly with head indicating a change in input head with output head variation

Figure 4 shows the relationship between isentropic head or enthalpy rise and flow rate for a screw compressor with assumed peak efficiency 85%. The performance line shows only a small reduction in flow with increase in head and no head limitation. The compressor efficiency varies as function of isentropic head or pressure ratio and flow rate. Compressor peak efficiency occurs at a head or pressure ratio corresponding to the built-in volume ratio (Figure 3a). Over-compression (Figure 3b) and under-compression (Figure 3c) will result in additional losses and reduced efficiency. However the fall-off of efficiency at lower head is less than that of a centrifugal compressor thanks to the input head reduction of screw compressors during over-compression.

Continuous off-design operation of screw compressors can be achieved by the use of variable geometry compressor hardware. A slide valve can adjust the amount of flow being compressed. The single one-dimensional head/flow characteristic shown in Figure 4 can now be replaced a two-dimensional area of possible compressor operation that can be achieved with different slide valve positions. By connecting points of equal compressor efficiency for the different slide valve positions a performance map with efficiency islands can be obtained. Figure 5 shows a compressor map obtained in that fashion. The screw compressor map has only a capacity boundary (fully open slide valve position). The dashed lines in this figure are the normalized efficiency islands. They show the relative change in efficiency at off-design conditions.

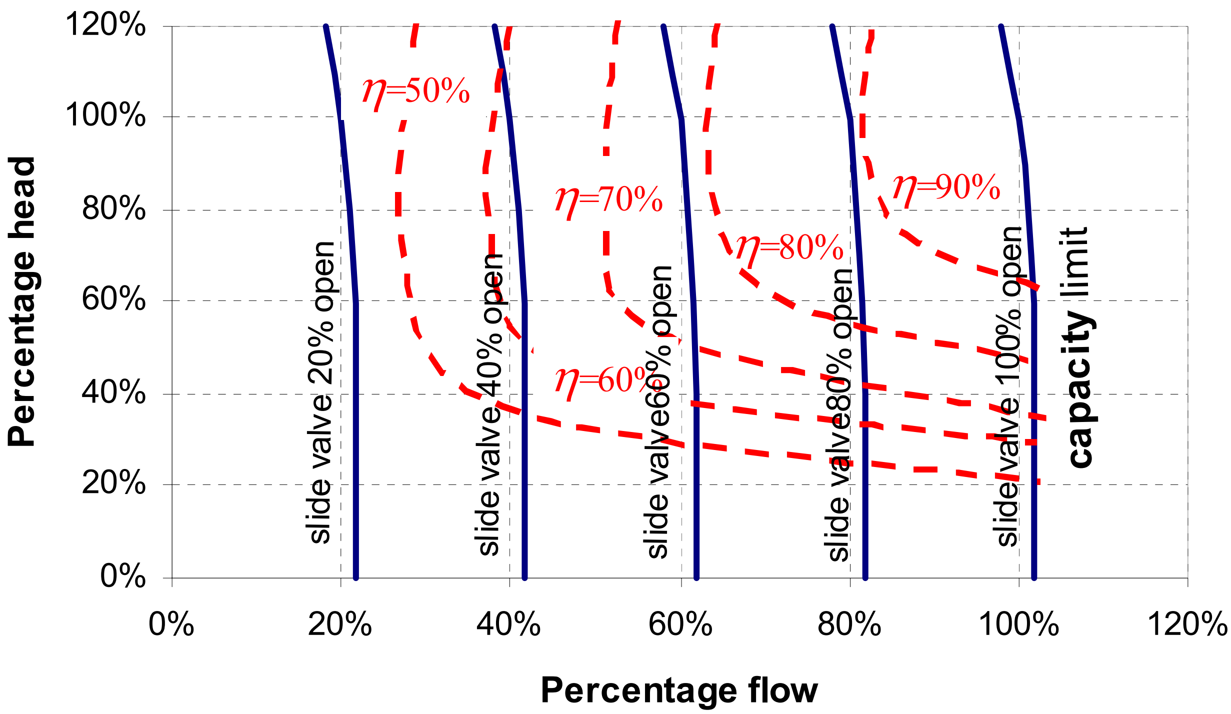

Figure 5. Fixed speed screw compressor performance map with slide valve capacity control

4. PART-LOAD COMPARISON OF VARIOUS SCREW AND CENTRIFUGAL COMPRESSORS

Using the process outlined above performance maps can be developed for various types of compressors and by comparing the shape and location of the efficiency islands we can determine the relative merit of various compressor concepts for given applications. This has been done for the following 5 compressor concepts:
1. Fixed speed vaned-diffuser centrifugal compressor with inlet guide vanes
2. Fixed speed vaned-diffuser centrifugal compressor with variable diffuser geometry
3. Fixed-speed oil-flooded screw compressor with slide valve
4. Variable speed low-oil screw compressor .
5. Variable speed vaned-diffuser centrifugal compressor with variable inlet guide vanes

The relative part-load efficiencies of these compression concepts are shown in Figure 6. The 95% efficiency islands for each of these concepts are drawn for each of these concepts. In other words, efficiency is better than 95% of design efficiency for head/flow combinations inside the islands (meaning between the full-load design point and the island contours) shown on the map.

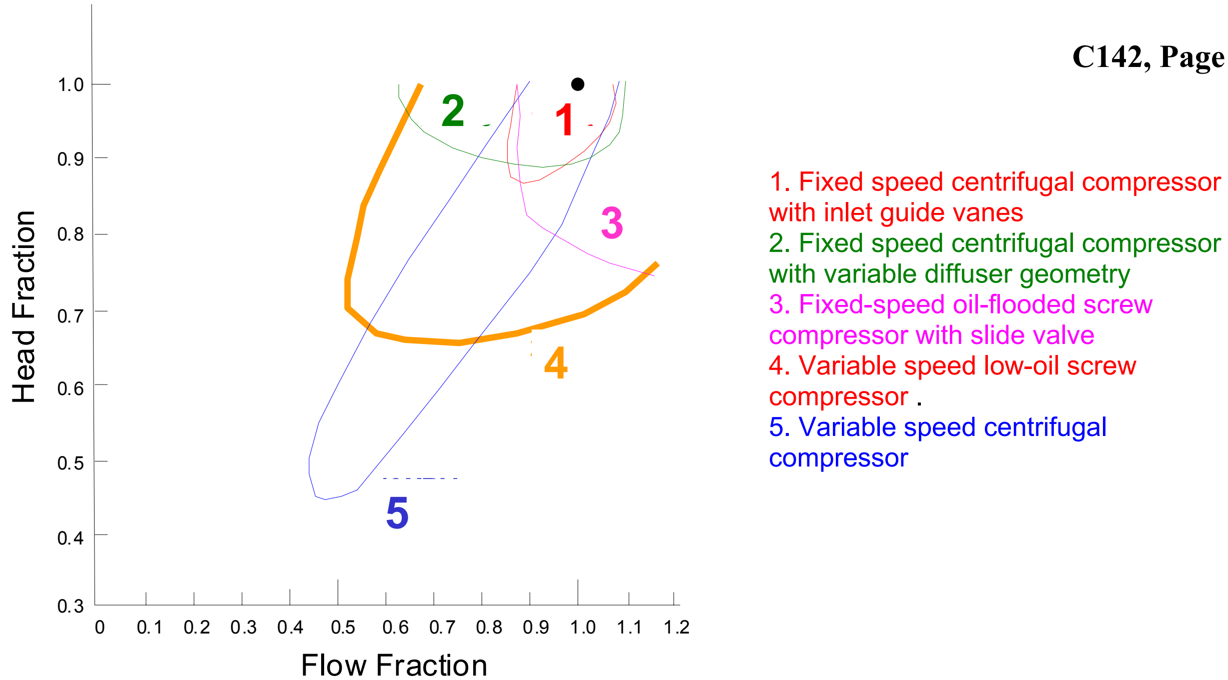

Figure 6. Comparison of map areas with relative part-load efficiencies > 95% for various centrifugal and screw compression concepts

5. SUMMARY AND CONCLUSIONS

1. For fixed speed operation screw compressor efficiency reduces less at lower head fraction than centrifugal compressors.
2. Variable diffuser geometry centrifugal compressors allow more efficient high-lift low-flow operation than fixed geometry diffuser centrifugal compressors.
3. The low-oil, variable-speed screw compressor improves part-load efficiency at lower flow rates than a fixed-speed oil-flooded slide-valve-controlled screw compressor.
4. Screw compressor build-in volume ratio reduces efficiency at lower heads whether slide-valve or variable speed controlled.
5. At reduced head levels centrifugal compressors benefit more from variable speed operation than screw compressors.

NOMENCLATURE

Symbols

c	velocity
H	head
h	enthalpy
P	pressure
s	entropy
T	temperature
u	impeller speed
V	specific volume
η	efficiency

subscripts

1	inlet
2	exit
is	isentropic
in	input
act	actual
θ	tangential

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