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THE EFFECT OF IMPELLER INLET ANNULAR TURNING VANES ON MULTISTAGE CENTRIFUGAL COMPRESSOR PERFORMANCE

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BREAK

ABSTRACT

The flow path of multistage centrifugal compressors is characterized by two 180-degree turns per stage: the inlet turning bend connecting the radial inflow return channel with the radial outflow impeller and the cross-over bend connecting the radial outflow diffuser with the return channel. Due to higher flow velocity and larger width to turning radius ratio, the turning losses are substantially higher in the inlet bend than in the return channel bend. Performance measurements were taken using different annular through-flow turning vane arrangements designed to reduce the inlet turning losses and increase the overall efficiency of the multistage centrifugal compressor. The experiments have shown consistent efficiency gains with corresponding capacity increases by adding multiple annular turning vanes in the inlet bend. The performance improvement potential of the vanes depends strongly on the positioning of these vanes in the flow passage. Based on these results, an empirical turning loss model was developed with the capability to predict the performance improvement achievable with correctly positioned single or multiple turning vanes in the impeller inlet bend area.

NOMENCLATURE

b	flow passage width
C	turning loss coefficient constant
C_f	friction factor
D_{hyd}	hydraulic diameter
L_v	vane equivalent passage length
lc	loss coefficient
N	number of turning vanes
P	static pressure
R_v	turning vane radius of curvature
R_c	turn mean radius of curvature
Re	Reynolds number
R_h	hub radius of curvature
RPM	compressor rotational speed (RPM)
R_s	shroud radius of curvature
V_m	flow meridional velocity
α	turning loss coefficient exponent
ρ	density
ΔH_v	annular turning vane head loss
ΔH_{sf}	skin friction head loss
ΔH_{turn}	overall turning head loss
H_i	stage input head
H	adiabatic head

INTRODUCTION

The axial length of a multistage compressor is a compromise between conflicting mechanical and aerodynamic requirements. A short drive shaft is desired from a mechanical point of view in order to prevent critical speed problems. Ample axial stage spacing is desired from an aerodynamic point of view to allow gently turning gas passages, resulting in maximum machine efficiency. The larger the number of stages mounted on a single shaft, the stronger this conflict becomes.

This problem is probably most pronounced for the application of directly-coupled, electric-motor-driven, multistage industrial blowers and exhausters. The typical 2:1 pressure ratio requirement for blowers and exhausters in the capacity range of 2 to 20 m^3/s can result in machines with up to 10 stages in series. Figure 1 shows a cut-away view of a five-stage multistage centrifugal blower directly coupled to an electrical

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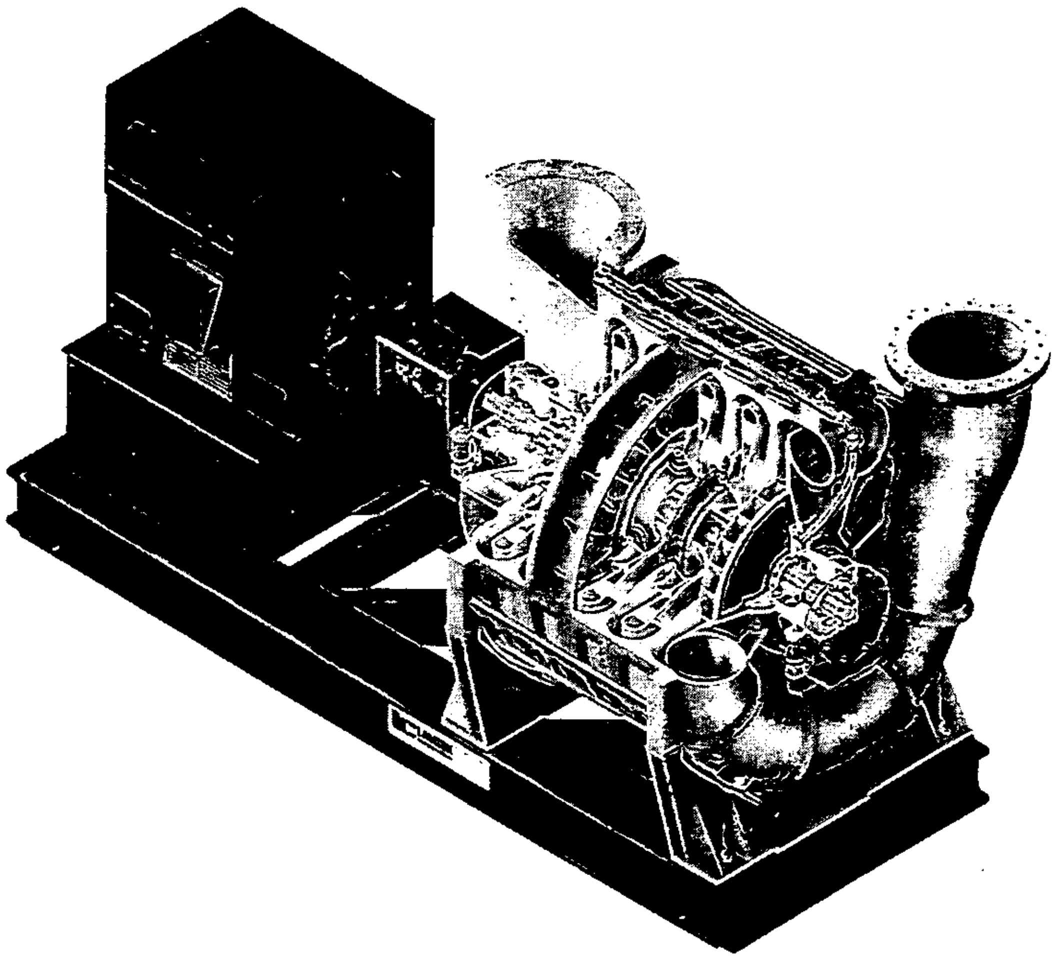


Fig. 1 Cut-away view of a five-stage centrifugal compressor

motor. Axial-stage spacing is obviously at a premium for those multistage machines. As a consequence, extremely an sharp turn occur in the aerodynamic gas passage at the inlet of the impeller. The resulting drop in machine efficiency was commonly thought to be inevitable.

For a given axial stage spacing, the compressor designer may choose a sharply curved but wide turn with low meridional velocity causing a high turning loss due to the large value of the turning loss coefficient. The alternative is a somewhat gentler narrow turn at much higher through-flow velocity causing a high turning loss despite the somewhat lower turning loss coefficient due to the larger value of the through-flow velocity (see Figure 2). A splitter turning vane, known as "baffle ring" in the blower industry, is often installed close to the sharp curvature of the bend to reduce flow turning losses and improve overall machine efficiency.

The "cascade" analogy of a compressor inlet bend is an elbow with a rectilinear cross section. Extensive studies have been carried out on flow turning losses in such elbows (e.g. Madison and Parker, 1936, McLellan and Barlett, 1941, Weske, 1943, Ahmed and Brunditt, 1969, Wallis, 1983). To reduce the turning flow losses for given elbow dimensions, splitter turning vanes have been inserted. Rules have been established which describe the optimum location of such turning vanes.

The object of the study described in this paper was to determine the benefit of inserting splitter turning vanes in the impeller inlet bend of a multistage centrifugal compressor. It was found, experimentally, that the location of multiple annular turning vanes is critical. For example, in an early phase of the program, it was observed that evenly spaced annular turning vanes at the impeller inlet actually reduced overall blower efficiency. However, properly positioned annular turning vanes in the impeller inlet bend that follow spacing arrangements similar to those recommended for bends with a rectilinear cross section improved the compressor performance dramatically. The concept of multiple annular turning vane insertion was applied to two different types of blower designs: one with a

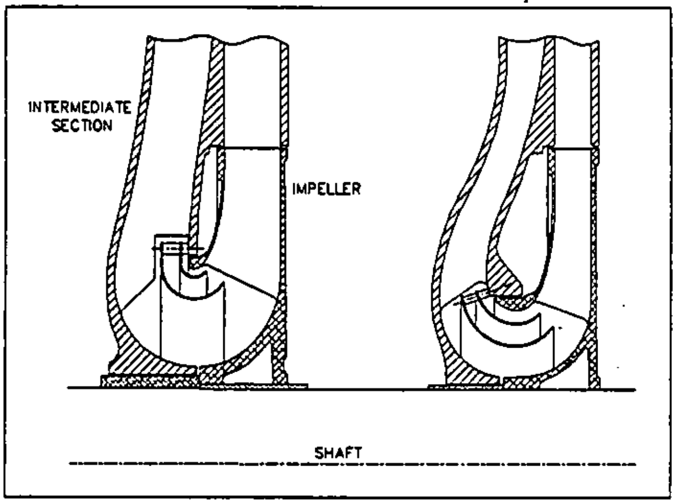


Fig. 2 Comparison between two impeller inlet turn design in a multistage centrifugal compressor

very sharp turn and a low through-flow velocity in the bend and one with a milder turn but higher through-flow velocity. As expected, the benefit of the multiple annular turning vanes is largest for the low through-flow/sharp turn blower design. Based on the results of this investigation the preferred design uses a wide through-flow velocity turn with a multiple annular turning vane arrangement to reduce the turning loss coefficient (Brasz, 1994).

1. EXPERIMENTAL INVESTIGATION

1.1 Positioning of the ennuler turning venes

The stationary turning vanes have a toroidal contour and a camber engle of approximately 150°. The uniform thickness vanes are located at the impeller inlet and mounted on the inlet head for the first stage or on the intermediate sections for the subsequent stages. A typical arrangement with two annular turning vanes is shown in Fig.3. A pressure loss occurs in the inlet turning bend because of the pressure gradient across the turn which causes a difference in flow velocity between the inside and the outside of the turn. The flow velocity is higher at the inside of the turn. Annular turning vanes create a more even flow of gas into the impeller, thus reducing losses while preventing flow separation.

The presence of several annular turning vanes in the inlet bend results in several subchannels for the gas flow. Turning losses ere proportional to the maximum pressure difference across the flow passage. It is therefore desirable to reduce this pressure difference. The latter can be derived from the radial equilibrium equation:

$$\frac{\partial P}{\partial n} = \rho \frac{V_m^2}{R_c} \tag{1}$$

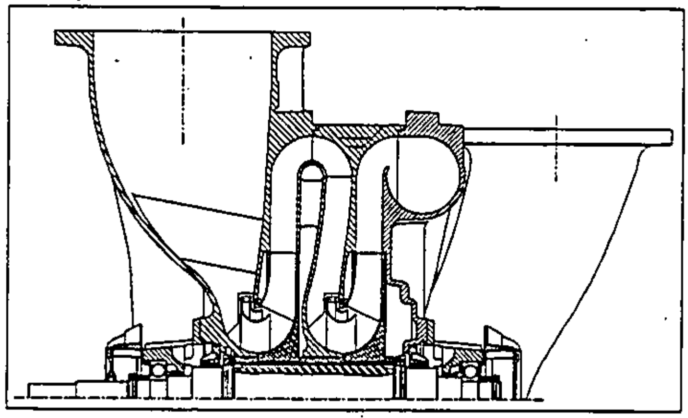


Fig. 3 Typical two-annular turning ring arrangement on a multistage blower. Lamson two-stage 2000 blowar.

where R_c is the turn mean radius of curvature, P the static pressure, $\hat{n}$ the unit vector normal to the bend, ρ the gas density and V_m the flow meridional velocity. Hence, the maximum radial pressure difference in a turn with width b, assuming a constant velocity profile across the flow channel, is:

$$\Delta P = \rho\left(\frac{b}{R}\right)V_m^2 \tag{2}$$

Two strategies can be adopted to position the annular turning vanes. The turning vanes may either form subchannels with equal cross sectional area or with equal pressure loss. Applying the rule of equal cross sectional area, for one annular turning vane, results in a radius of curvature R_v for the vane of:

$$R_v = \sqrt{\frac{R_h^2 + R_s^2}{2}} \tag{3}$$

where R_h and R_s are the hub and the shroud radii of curvature, respectively. The ratio (b/R) is larger in the subchannel on the shroud side inducing higher losses than in the subchannel on the hub side. This results in an uneven flow into the impeller and degraded performances.

The radial pressure difference will be minimum if it is the same in all subchannels. In order to achieve equal losses in two subchannels with one annular turning vane, the positioning of the ring must be such that:

$$R_v = \sqrt{R_s \times R_h} \tag{4}$$

This formula can be generalized to the case of N annular turning vanes:

$$
\begin{aligned}
R_{v_1} &= \sqrt[N+1]{R_s^N \times R_h} \\
R_{v_2} &= \sqrt[N+1]{R_s^{N-1} \times R_h^2} \\
&\dots \\
R_{v_K} &= \sqrt[N+1]{R_s^{N-K} \times R_h^K} \\
&\dots \\
R_{v_N} &= \sqrt[N+1]{R_s \times R_h^N}
\end{aligned}
\tag{5}
$$

Note that similar design rules have been derived and used for circular pipes (Ito and Imai, 1966, Wallis, 1983).

The location of the vane leading edge must be such that the flow incidence on the vane is minimal. In the centrifugal blower design considered here, the flow coming out of the return channel is approximately normal to the blower axis. Therefore, the inlet of the annular turning vane is in the radial direction, normal to the shaft.

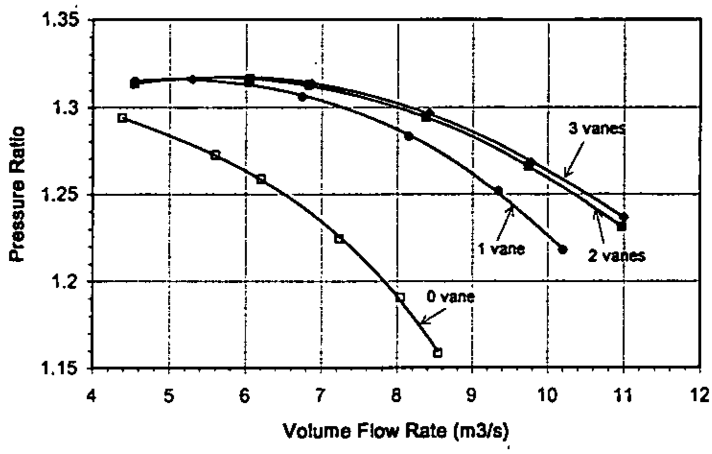
1.2 Test facility and blower

The tests were carried out at Lamson test laboratory in accordance with ASME PTC 10 code. The multistage centrifugal blower under investigation belonged to Lamson's 2000 series. The 2000 series features a capacity range from 2 to 12 m^3/s and a pressure ratio of up to 2, with the maximum number of stages. A two-stage blower was assembled using the most radial impellers designed for the 2000 series. Its cross section is shown in Fig. 3. The inlet design flow was 8 m^3/s and the design pressure ratio 1.3. The parameters measured during the experiments included inlet volume flow rate, inlet and discharge pressures and temperatures, and power consumption. On this particular blower, the radius of curvature of the impeller inlet was sharp while the flow passage area was fairly large.

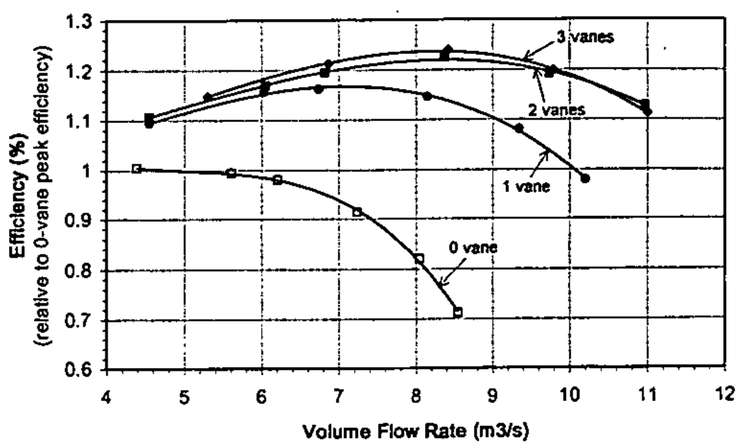
1.3 Results

The annular turning vane location yielding equal pressure losses across the subchannels was selected because it was expected to give the highest performance gains. Therefore, the vanes were located according to the equations (5) above. Several configurations were tested with varying numbers of annular turning vanes. For reference purpose, the first test was on a blower without turning vanes. Subsequent tests were conducted with one, two and three turning vanes.

The test results are plotted in Fig. 4 in terms of pressure ratio and efficiency against volumetric flow rate. The blower performance improved drastically with the addition of annular turning vanes. The largest gain was between zero and one annular turning vane: the increase in peak efficiency was 14%. The addition of a second annular turning vane resulted in a peak efficiency increase over one turning vane of 4% and the presence of a third turning vane further increased the peak efficiency by 1.5%. At the same time, the range of volumetric flow rates increased by 25% with the presence of one annular turning vane over no annular turning vane. This increase in flow was accompanied by a substantial increase in discharge pressure. Adding a second and a third annular turning vane further contributed to a larger flow range and higher pressure ratios.



a) Pressure ratio



b) Efficiency

Fig. 4 Two-stage 2000 series blower performance with various number of turning vanes

2. ONE DIMENSIONAL LOSS MODEL

2.1 Loss model

A simple mean streamline model was developed to determine the inlet turning losses and the potential performance improvements with the presence of annular turning vanes at the impeller inlet turn. The aerodynamic losses in the bend that were taken into account in this simple loss model were the turning loss and the skin friction loss.

The turning loss is proportional to the product of the mean through-flow velocity squared times a loss coefficient. As stated earlier, this loss coefficient is proportional to the maximum pressure difference across the flow channel, which is in turn proportional to the ratio of the passage width to the turn mean radius of curvature (see Eq. (1)). A general expression for the pressure loss in the bend is therefore:

$$\Delta P = C \times \left(\frac{b}{R_c}\right)^{\alpha} \times \rho \times V_m^2 \tag{6}$$

where C and α are two constants to be determined from the experimental data.

The skin friction loss is approximated by a formula by Jansen (1967) intended for impellers. The flow is assumed to be equivalent to a fully developed turbulent flow in a pipe of circular cross section with a diameter equal to the average hydraulic diameter D_{hyd} of the subchannel on the shroud side and a length equal to the average length of the same flow passage. This approximate method has been widely used for non-circular pipes and is extended to the toroidal flow passage studied here. The expression for the friction loss is:

$$\Delta H_{sf} = 2C_f \times \left(\frac{L_v}{D_{hyd}}\right) \times \frac{V_m^2}{2} \tag{7}$$

where C_f is the friction factor for a pipe of diameter D_{hyd}. In the present case, the average equivalent hydraulic diameter is:

$$D_{hyd} = \frac{4 \times Area}{Perimeter} = \frac{4b\pi D_0}{2\pi D_0} = 2b \tag{8}$$

and the passage length: $L_v = \pi\left(R_v - \frac{b}{2}\right) \times \frac{\theta}{180} + l_e$

where θ is the camber angle of the annular turning vane (in degrees) and l_e is the length of the straight passage before the curved annular turning vane. The friction factor is calculated as:

$$C_f = \frac{0.0412}{\left(\mathrm{Re}_{D_{hyd}}\right)^{.1925}} \tag{9}$$

where Re_{Dhyd} is the Reynolds number based on the flow velocity at the impeller inlet and the hydraulic diameter (Jansen, 1967).

A useful tool that was utilized to qualify the loss correlation is a performance prediction computer program developed at Lamson. This program predicts the performance of a multistage blower with given number of stages, impeller configuration and inlet conditions. Non-dimensional head and efficiency curves, derived from test data for each blower series and impeller type, constitute a data base for the program together with necessary geometrical parameters. For a given inlet volumetric flow rate into the stage, each stage is assumed to produce the same head with the same efficiency. This assumption is legitimate for low Mach number flows where the compressibility effect is negligible. The program stacks up stages while updating the gas parameters between each stage. Varying with the blower radial inlet head design, a pressure drop is calculated to simulate the loss due to the flow asymmetry into the first stage. This performance prediction model has proven to be very accurate for all frame sizes.

The approach adopted to predict the blower performances with varying numbers of turning vanes consisted of considering the effects of the vanes at various levels, i.e. at the component, stage and overall machine levels. The component of interest was the annular turning vane assembly. The sum of the turning loss and the skin friction loss was:

$$\Delta H_v = \Delta H_{turn} + \Delta H_{sf} = \left[C \times \left(\frac{b}{R}\right)^{\alpha} + C_f \times \left(\frac{L_v}{D_{hyd}}\right)\right] \times V_m^2 \tag{10}$$

At the stage level, the standard two annular turning vane head and efficiency curves were modified to reflect the different configurations. The assumption here was that the number of annular turning vanes only affected the losses in the turn while all other stage losses remained the same. The difference in stage adiabatic heads between the configurations with two annular turning vanes and n annular turning vanes was therefore equal to the difference between the losses in the turn for these two configurations, calculated as:

$$H_{nv} - H_{2v} = \Delta H_{2v} - \Delta H_{nv} \tag{11}$$

for n = 0, 1 and 3.

Using the expression for the losses in the turn, the new stage adiabatic head was:

$$H_{nv} = H_{2v} - \left[C \times \left(\frac{b}{R}\right)_2^{\alpha} + C_{f2} \times \left(\frac{L_v}{D_{hyd}}\right)_2 - C \times \left(\frac{b}{R}\right)_n^{\alpha} - C_{f_n} \times \left(\frac{L_v}{D_{hyd}}\right)_n\right] \times V_m^2 \tag{12}$$

V_m is the average meridional velocity at the inlet of the first impeller. The stage input head H_i, defined as the ratio of the adiabatic head over the adiabatic efficiency was assumed to be constant and used to calculate the new stage efficiency as:

$$\eta_{nv} = \frac{H_{nv}}{H_i} \tag{13}$$

$$\text{where} \quad H_I = \frac{H_{2v}}{\eta_{2v}} \tag{14}$$

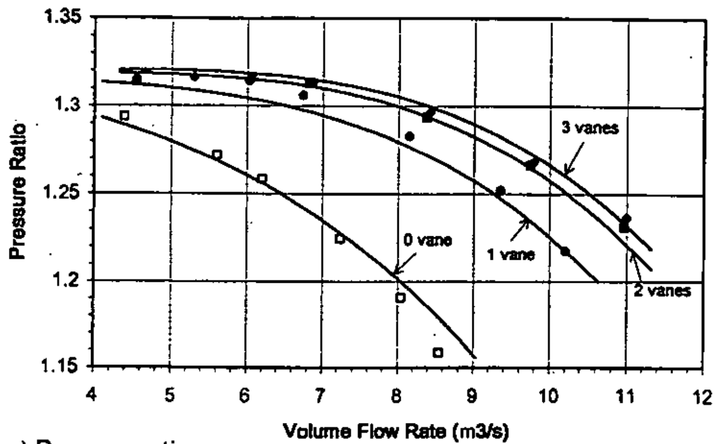
Finally, the machine performance was calculated with the prediction program by stacking up stages. The coefficients C and α were adjusted so that the predicted pressure and efficiency curves would best match the experimental data.

2.2 Application of the loss model to the Lamson 2000 series blower

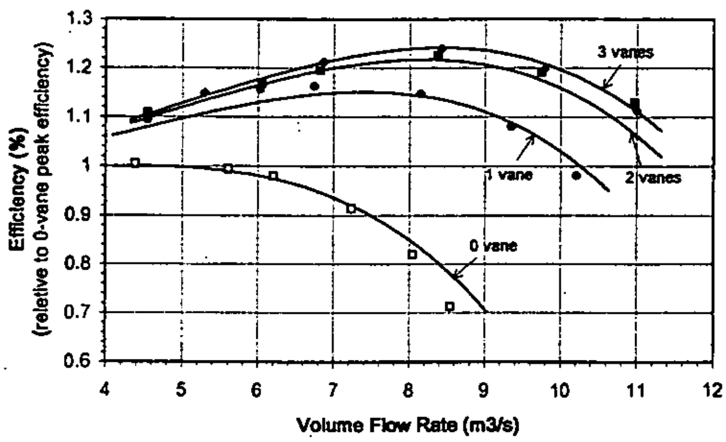
The model described above was applied to the 2000 series blower. Based on the test data, the correlation for the turning loss was:

$$\Delta H_{turn} = 0.2 \times \left(\frac{b}{R_c}\right)^{3.2} \times V_m^2$$

The curves produced with the final coefficients are plotted in Fig. 5. Remarkably, the trends observed experimentally were very closely reproduced by the loss model: the difference between zero and one annular turning vane is the largest while the benefit of adding a third annular turning vane over two is the smallest. The maximum difference between the experimental and the predicted pressure curves is less than 5% for the cases of one, two and three annular turning vanes and less than 15% for the zero-annular turning vane configuration. The efficiency was also very well predicted with a maximum difference between predicted and experimental data of 10%.



a) Pressure ratio



b) Efficiency

Fig. 5 Comparison between predictions and experimental data. Two-stage 2000 series blower.
Experimental data: □ no vane, ● 1 vane, ■ 2 vanes, ◆ 3 vanes

The passage width to radius of curvature ratios and the two loss coefficients for zero to three annular turning vanes are given in Table 1 together with similar values for the 1850 series blower which will be introduced later. Note that the skin friction coefficient is the average value over the blower volumetric flow range.

The skin friction head loss was less than 1% of the stage total head. Its importance relative to the turning loss was less than 0.5% for the zero annular turning vane configuration and increased with increasing number of annular turning vanes to approximately 25% for the case of three annular turning vanes.

Table 1 Passage width to radius of curvature ratios and loss coefficients for various numbers of turning vanes.
lc turn = $0.2\,(b/R_c)^{3.2}$, lc skin friction = $C_f\,(L/D_h)$

number of vanes	2000 series			1850 series		
	b/R	lc turn	lc skin friction	b/R	lc turn	lc skin friction
0	1.76	1.221	0.0034	1	0.2	0.0054
1	1.194	0.353	0.0042	0.536	0.0272	0.0098
2	0.858	0.123	0.0087	0.362	0.0077	0.0144
3	0.662	0.053	0.0119	0.273	0.0031	0.0191

2.3 Application of the loss model to Lamson 1850 series blower

The loss model was applied to a four-stage blower from Lamson's 1850 series to validate the model as well as to see the benefit of annular turning vanes for a blower with a larger mean radius of curvature at the impeller inlet turn. The blower with a two annular turning vane arrangement is shown in Fig. 6. The design flow was 5 m^3/s and the pressure ratio was 1.6. The two-annular turning vane configuration is the standard predicted by the performance prediction program. The performances with zero, one and three annular turning vanes were calculated using equations (12) and (13), and compared with the test data available (zero and two annular turning vanes).

The predicted pressure ratio and efficiency curves are plotted with the experimental data points in Fig. 7. The use of annular turning vanes improved both the pressure and the efficiency. Interestingly, the performances with one or two annular turning vanes were almost identical and the addition of

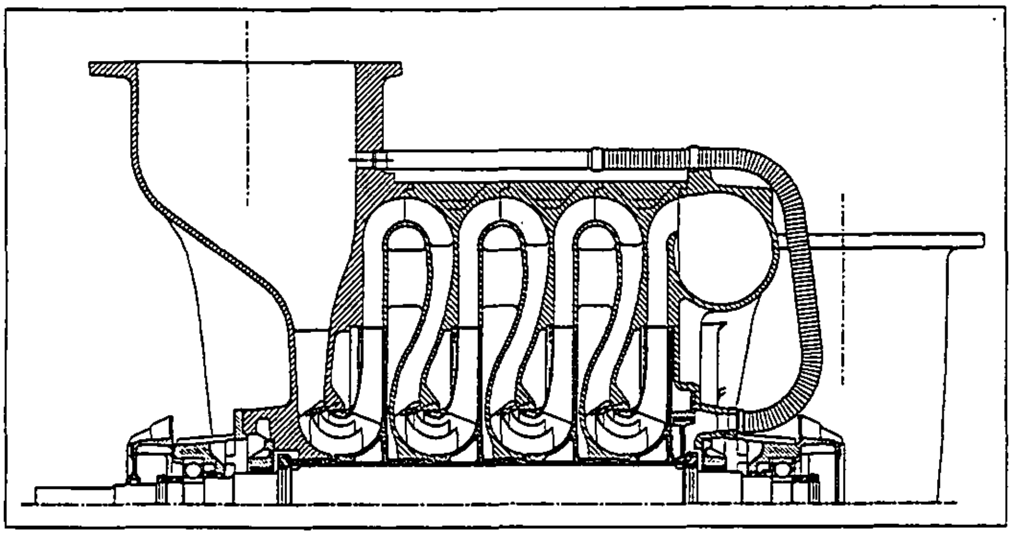


Fig. 6 Multiple turning vane arrangement on Lamson four-stage 1850 series blower

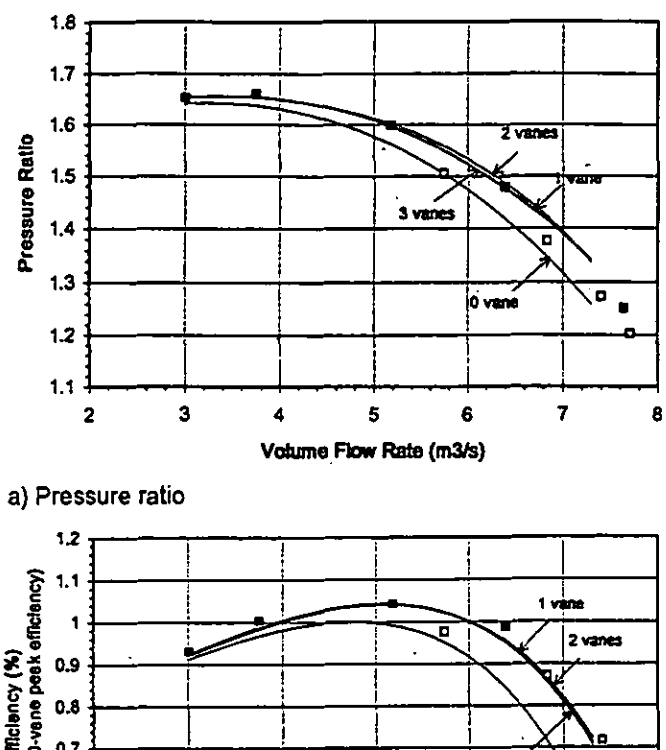
a third annular turning vane actually slightly reduced the blower performance. Referring to Table 1, it can be observed that, indeed, the sum of the turning loss coefficient and the skin friction coefficient in the case of three annular turning vanes was almost equal to that for two annular turning vanes. It can also be noted that the skin friction loss became the predominant loss mechanism for two and three vanes. The conflict between smaller turning loss and larger friction loss with increasing number of annular turning vanes becomes clear and, along with it, the optimum number of annular turning vanes. The predictions are close to the test data available for zero and two annular turning vanes, though the efficiency with no annular turning vane is somewhat underpredicted.

It is of interest to note the similarity of these results with those in the discussion of the paper by Madison and Parker (1936). In that study, the effect of various numbers of splitters (one to three) in square section elbows was plotted versus the curve ratio, defined as the ratio of the turn inside radius of curvature over the outside radius of curvature. It was found that the optimum number of splitters in the elbow was not necessarily the highest number. For some curve ratios, three splitters resulted in larger losses than one or two splitters.

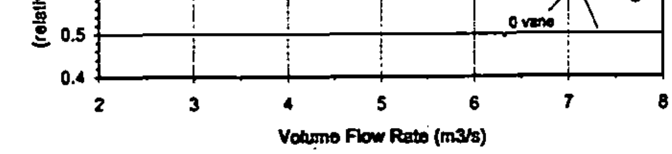
2.4 Accuracy and validity of the model

In view of the two examples studied in this paper, it can be stated that the accuracy of the model will be better when the ratio of passage width to mean radius of curvature is large. In that case, the turning loss dominates other loss mechanisms. In the first example of the 2000 series blower, the ratio of passage width to mean radius of curvature varied between 0.66 and 1.76 and the accuracy was within 15%.

For ratios of passage width to mean radius of curvature smaller than 0.6, the skin friction produces a major part of the global loss in the turn. It is therefore important to have a good



a) Pressure ratio



b) Efficiency

Fig. 7 **Comparison between predictions and experimental data. Four-stage 1850 series blower. Experimental date: □ no vane, ■ 2 vanes** model to predict the skin friction loss. Further study to evaluate this loss would be of interest, though it was not the focus of this paper.

2.5 Comparison with other turning loss correlations:

The expression for the turning loss was compared with existing models for losses in pipe and duct bends. Loss coefficients are tabulated in the ASHRAE Hanbook of Fundamentals (1972) for the total pressure loss due to elbows of rectangular section. For closest analogy with the turn investigated in this paper, the largest aspect ratio width to height available is considered. The coefficients for the 90° degree elbow turn are $C = 0.19$ and $\alpha = 2.37$. The loss in an angle of 150° was calculated as 1.4 times the loss in the 90° elbow, based on results by Madison and Parker (1936) for rectangular elbows. For large ratios passage width to mean radius of curvature, the turning loss in the annular turning vanes was larger than in the elbow. This means that the addition of annular turning vanes induced greater benefit. This is due to the fact that in addition to the reduction in the turning losses, the flow out of the annular turning vanes and into the impeller was more uniform, resulting in better performances.

CONCLUSIONS

Properly positioned multiple annular turning vanes in the inlet bend of narrowly spaced multistage compressors have been found to improve performance dramatically.

Since the effect of the multiple annular turning vane concept is a reduction of the turning loss coefficients, its benefit is largest for high flow applications. Therefore, multiple annular turning vanes allow increased capacity out of a given multistage blower design.

As a result of the effectiveness of the multiple annular turning vane concept, the preferred design approach for a return channel and inlet bend combination of a multistage blower is to reduce the through-flow velocity as much as possible in the return channel and to reduce the turning loss coefficient with turning vanes.

The reduction in turning flow loss found for large ratios of passage width to mean readius of curvature was larger than originally anticipated based on the published benefits of splitter vanes for rectilinear bends. This is due to the fact that annular turning vanes not only reduce losses in the bend, they also deliver a more uniform flow into the impeller causing its efficiency to increase.

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