PERFORMANCE SPECIFICATIONS

RDH fans

Attachment to Nicotra "Ventil" selection program
version 2.0.0 or following

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Air performance

Air performance ratings of RDH fans, as described by the “Ventil” selection program with Upgrade Package CP2.5, have been derived from performance tests made with installation type B, with free inlet and ducted outlet. These tests were carried out in the Nicotra laboratory, in accordance with the following standards: AMCA 210-99 (fig. 12), UNI 10531 (fig. 30 c and par. 29.2 f) and ISO 5801 (fig. 69 c and par. 30.2 f).

“Free-outlet” operation

When operating in installation type “A”, with free inlet and free outlet, the available static pressure from the fan, p_{SA} , is lower than when the fan is used with ducted outlet, and can be satisfactorily calculated subtracting, from the total pressure, an increased dynamic pressure, calculated by multiplying conventional dynamic pressure (based on average velocity) by a factor K_d shown below.
This dynamic pressure increase represents the effect of the airflow contraction produced by the cut-off plate and the absence of an outlet duct, which would act as a diffuser, allowing at least partial conversion of the excess of dynamic pressure into static pressure.

K_d - RDH
1.74

Power

Power curves shown on fan performance diagrams are impeller absorbed power. Power consumption from the fan bearings is calculated separately for each fan version.
Fan shaft power is given by the addition of impeller power and power used by the bearings.
In most cases, bearing power is small and often negligible when compared against impeller power, but becomes relatively more important with decreasing fan size and speed, and may be significant at the lower end of the size range.
Drive losses are not calculated.
Fan mechanical input power is a function of flow rate and speed, but doesn’t change between installation types “A” (with free inlet and free outlet) and “B” (with free inlet and ducted outlet).

Efficiency

Efficiency values shown on the diagrams are total impeller efficiency, with the fan operating with installation type “B” (η_{rB} according to ISO 5801 symbols), without considering bearing power losses, drive losses and, of course, motor power losses.
Impeller efficiency actually is, for a given fan size, a function also of fan speed, or, alternatively, of the Reynolds number Re.
Experimental measurements have shown, anyway, that within the fan speed range shown in the catalogue, the actual efficiency variation of RDH impellers is well within the allowed tolerances. Making the catalogue even more complicated to accurately show even this small variation was so considered unjustified.

Fan total efficiency, with installation type B, η_{aB} (including bearing power consumption), can be calculated from impeller total efficiency with the following formula:

$$\eta_{aB} = \eta_{rB} \cdot \frac{W_r}{(W_r + W_b)}$$

where:

η_{rB} is impeller total efficiency, with installation type B

η_{aB} is fan total efficiency, with installation type B

W_r is the power used by the fan impeller

W_b is the power dissipated by the fan bearings

Fan static efficiency is efficiency calculated using only fan static pressure (and not total pressure) to calculate the useful power. As such, it is more representative of the actual fan energy efficiency when it is used with installation type "A", without a duct on the outlet.

The fan static efficiency, referenced to shaft power, with installation type A, η_{SaA} , can be calculated with the following formula:

$$\eta_{SaA} = \eta_{rB} \cdot \frac{W_r}{(W_r + W_b)} \cdot \frac{p_{SA}}{p_{FB}} = \eta_{rB} \cdot \frac{W_r}{(W_r + W_b)} \cdot \frac{(p_{FB} - K_d \cdot p_{dB})}{p_{FB}}$$

where:

p_{FB} is the fan total pressure with installation type B (as shown in performance diagrams),

p_{SA} is the fan static pressure with installation type A,

K_d is the coefficient for calculation of fan static pressure with installation type A,

p_{dB} is the fan conventional dynamic pressure with installation type B.

The Ventil selection program automatically calculates static and total fan efficiency values, referenced to the actual shaft power, for the selected installation type and each fan version.

Sound Power Level

The noise ratings of RDH fans are calculated starting from sound power level measurements made in accordance with the AMCA 300-96 standard, fig. 2 (inlet side measurements) and fig. 3 (outlet side measurements).
The measured values have been converted to other operating speeds with a calculation procedure described in the BS 848 Part 2 standard, Appendix G. This procedure is a more general version of the procedure contained in AMCA 301-90 and is in accordance with the currently available (July 2001) draft version of the ISO 13348 standard, under development by the ISO TC 117 technical committee.

The Ventil selection program carries out integrally these calculations, and gives the best approximation of the results.

The LWS curves on the fan performance diagrams show the fan A-weighted Sound Power Level (Lws_7(A)) on the inlet side (inlet noise plus case radiated noise).

Outlet side sound power levels ($Lws_{4\text{-}Oct}$, $Lws(A)_4$ and Lws_4), calculated with the Ventil selection program, are values equivalent to in-duct sound power levels as measured according to ISO 5136, DIN 45635 Part 9, BS 848 Part 2 Chapter 6 or AMCA 330-97.
These sound power levels differ from outlet side values measured in accordance with the AMCA 300-96 standard because of the subtraction of the end reflection correction, given, for each octave band, by the following formula:

$$E_{Oct} = 10 \cdot \log_{10}\left[1 + \left(\frac{20 \cdot \sqrt{293.15}}{f_{Oct} \cdot L \cdot \sqrt{4 \cdot \pi}}\right)^{1.88}\right]$$

where

f_{Oct} is the centre frequency of each octave band (63Hz, 125 Hz ….)
L is the fan outlet side length in m.

The user should remember that the sound power level of a fan, as installed in practice, could be significantly higher then that measured in laboratory conditions.
Mechanical vibrations at the rotation frequencies of the fan and of the driving motor, and at the internal vibration frequencies of the same motor can easily radiate noise which is not actually produced by the fan, usually with narrow peaks at well defined frequencies. The mechanical reaction of the fan structure to induced vibrations is highly influenced by the stiffness of the base frame holding together fan and motor.
Air leakage through the connections, or turbulence produced by guards, diffuser grids or transition pieces can also significantly change the broadband noise spectrum, particularly at high frequency.
As a final note, the actual amplitude of the blade passing frequency peak, a pure tone of aerodynamic origin, can be changed by many decibels as an effect of the acoustic resonance properties of the duct or plenum connected to the fan.
In most cases, the broadband noise level increase, produced by a reasonable real-life installation, is kept within 2 dBW. Much more difficult may be the prediction of the noise increase produced by mechanical vibrations, as this depends on the mechanical characteristics of many other components (base frame, motor, pulleys and so on), and of the blade passing frequency tone, which depends on the acoustic properties of the duct system.

Normal operation area

The performance diagram of RDH fans is divided into three areas by two red lines, parallel to the constant efficiency lines. The two red lines separate the normal operating area of the fan (in the middle), from the stall region (top-left area) and from the low-pressure region (bottom-right area).
An appropriate selection of the fan size gives an operating point within the normal operation area of the fan, and, ideally, on or slightly right of the best efficiency line of the fan.
A fan size selection producing an operating point in the low pressure area of the diagram should preferably be avoided, because of both the low efficiency and the larger uncertainty of the fan performance (see also DIN 24166 on this subject). To rectify such a selection, a larger fan size or a twin fan should be used, or a forward curved ADH fan should be substituted to a similarly sized RDH backward inclined one.
A fan size selection with the operating point in the stall region should be carefully avoided. Any fan operating in these conditions is not only scarcely efficient, but also generates a fluctuating pressure and a high noise level in the lower frequency octaves, which gives little contribution to the A-weighted total noise level, but may be highly annoying. Such a fan choice can be easily rectified selecting a smaller fan size.
The performance curves of the RDH fans have been extended outside the normal operation range to assist with troubleshooting at system start-up time, but fan performance in these conditions is subject to increased uncertainty, also because of the influence of the air system connected to the fan.

NOTE FOR USERS OF VENTIL FROM VERSION 1.0.0 TO 1.0.3 WITH THE CP 2.5 (OR LATER) UPGRADE PACKAGE

The RDH data archive files and the Nicotra.dll v. 2.0.0 file included within the CP 2.5 (or later) package are already capable of identifying a selection outside of the normal operation area (users of the DLL alone should look at the new version 2.0.0 user manual). The selection program Ventil of the said versions, anyway, is not yet capable of either showing the borders on the diagrams, or marking a selection made outside of them.
In case of doubt, a manual check can be done with the procedure described in the “Fast size selection procedure“ paragraph.
Starting from version 2.0.0, Ventil can show the normal operation area borders, and will clearly mark any selection made outside these borders.

Tolerances

RDH fans of the sizes from 180 to 315 have air performance, and sound power levels, as measured according to the AMCA 300-96 standard, within the tolerances allowed by the DIN 24166 standard for Class 2.
RDH fans of the sizes from 355 upward operate within the performance tolerances allowed by the same standard for Class 1.
Fan performance in the stall region is not guaranteed.

Performance of twin fan units G2

The performance of twin fan units, identified by the “G2” prefix, is calculated, starting from that in the corresponding operating point of a single fan, with the following formulas:

- pressure :	Pb = P x 1
- volume flow rate :	Qb = Q x 2
- impeller power :	Wb = W x 2,15
- fan speed :	Nb = N x 1,05
- Lws :	Lwsb = Lws + 3 dB

Quick size selection procedure

The faster way to identify the most appropriate fan size is to calculate the dimensional parabolic constant of the required operating point, K_P , defined as

$$K_P = \frac{p_{FB}}{Q^2} \cdot \frac{1.2}{\rho}$$

where

p_{FB} is the Fan Total Pressure in Pa,
Q is the Volume Flow rate in m^3/s, of the required operating point, and
ρ is the Air Density (1.2 kg/m^3 in standard conditions).

As a following step, the "K_{EtaOpt}" column should be searched, looking for the smaller value equal or larger than the calculated value. The fan size on the same level in the leftmost column is the first choice for a single fan.
To select a twin fan, only half of the total flow rate shall be used to calculate the parabolic constant K_{EtaOpt} .

Dimensional constant K_P [$Pa_t/(m^3/s)^2$]			
RDH	**KSx**	**K_{EtaOpt}**	**KDx**
180	18962	6668	1321
200	12042	3813	861
225	7471	2244	471
250	3315	1216	310
280	1635	773	234
315	1097	482	132
355	616	299	76.0
400	391	186	46.4
450	236	116	31.6
500	170	76.0	20.1
560	109	48.3	12.2
630	48.3	30.2	7.63
710	30.1	18.7	5.04
800	21.6	11.6	2.93
900	12.2	7.24	2.06
1000	8.49	4.75	1.21

The columns KSx and KDx contain the values of the parabolic constant K_P which, for each fan size, mark respectively the left and right limits of the normal operation region.
Values of K_P higher than KSx mean operation in the stall area, while K_P lower than KDx means operation in the low pressure area.
The use of the fan outside the normal operation range should be avoided, and particularly in the stall region.
If the K_P value is larger than KSx, the problem can be solved reducing the fan size.
If the K_P value is smaller than KDx, a better selection can be achieved using a larger size or a twin fan.

Motor selection

As already explained in the "Power" paragraph, the fan shaft power can be calculated adding together impeller power, W_r, and bearing power, W_b.
The minimum motor power required to drive the fan can be calculated multiplying this fan shaft power by a coefficient (function of the shaft power value) which includes both the belt drive power loss, and a reasonable safety margin. This safety margin covers any small change in the operating point or in the actual fan speed, which may be due to a motor speed or to a drive ratio slightly different from their design values.

$$W_{Tot} = (W_r + W_b)$$

$$W_{Mot} \geq W_{Tot} \cdot K_w$$

where

W_{Tot} is the fan shaft power
K_W is the motor selection coefficient

For RDH fans

$K_W = 1.25$ if $W_{Tot} < 0.75$ kW
$K_W = 1.15$ if 0.75 kW $\leq W_{Tot} < 10$ kW
$K_W = 1.12$ if $W_{Tot} \geq 10$ kW

The safety coefficients may be reduced if the actual operating point is precisely known, and the belt drive loss can be accurately calculated.

With motors larger that 7.5 kW the use of a star/delta (Y/Δ) starter or, alternatively, of a soft starter is highly recommended, to keep down starting current and to reduce mechanical stress.