ISSN 1392 - 1207. MECHANIKA. 2011. 17(2): 162-167

Effects of VNT and IGV association on turbocharger centrifugal compressor performances

L. Izidi*, A. Liazid**
Laboratoire LTE, ENSET d'Oran, BP 1523 El-M'naouer Oran 31100 Algérie, E-mail: lh_izidi@yahoo.fr
***Laboratoire LTE, ENSET d'Oran, BP 1523 El-M'naouer Oran 31100 Algérie, E-mail: ab_liazid@hotmail.com*

crossref http://dx.doi.org/10.5755/j01.mech.17.2.332

Nomenclature

C - absolute velocity, m/s; C_p - specific heat at constant pressure, kJ/kg K, D - impeller diameter, m; GV - opening ratio of turbine nozzle blades, %; IGV - Inlet Guide Vane; $\dot{m}$ - air mass flow rate, kg/s; M - mach number; N - rotational speed, rpm; P - pressure, Pa; r - radius, m; T - temperature, K; VNT - variable nozzle turbine; W - relative velocity, m/s; α_w - prewhirl angle, deg; ΔH_{SF} - skin friction losses; ΔH_P - power losses; ΔH_L - leakage losses; ΔH_{DF} - disc friction losses; ΔH_{BL} - blade loading losses; ΔH_{INC} - incidence losses; ΔH_{Dif} - diffuser losses; ΔH_V - volute losses; η - efficiency; π - pressure ratio.

Subscripts

1 - impeller inlet; 2 - impeller exit; a - axial component; c - compressor; i - inlet; is - isentropic; m - mean; o - outlet; r - root, radius, ratio; s - specific; t - tip; tc - turbocharger; w - whirl component, relative.

1. Introduction

Automotive turbochargers require small dimension stages able to supply the engine fresh mixture or air inquire over a wide speed and load range, McCutcheon and Brown [1], Capobianco and Gambarotta [2]. Radial turbines with nozzle guide vanes have found a wide application in diesel engine turbochargers and a lesser extent in small gas turbine engines. Matching the turbocharger geometry to engine operating conditions normally ensures good engine performances. The benefits of variable geometry turbochargers are described by McCutcheon and Brown [1] and Okazaki et al. [3]. Several configurations were tested and evaluated such as movable side walls volute and variable geometry stators, Capobianco and Gambarotta [2]. One of the most efficient configurations consists in pivoting the turbine stator blades to modify the flow incidence and therefore the nozzle section area. The influence of a variable guide vane nozzle on the design parameters of a radial turbine stage is studied by Binder et al. [4]. The authors found that an important variation around the nominal operating design geometry disturbs the performance characteristics and the initial design parameters are not conserved. Consequently, the nominal operating design approach is clearly not sufficient to get an adapted geometry for a large operating range. They suggest extending this approach to a new concept which takes into account a large operating range of the turbine. But it is still difficult for the designer to take into account the complexity of variable geometry stages from the very first steps of the design. The compressor performances are important in turbocharging field since they determine the engine air supply. This paper deals with the simultaneous influence of VNT and IGV techniques on operating performance characteristics of turbocharger centrifugal compressor. The study identifies this influence in actual operating conditions and examines the interest of a recent prewhirl design developed by Najjar and Akeel [5]. An experimental investigation of VNT influence was carried out and a prediction model for centrifugal compressor performances integrating a suggested IVG was developed to achieve the purpose.

2. Experimental setup

Experimental measurements were carried out on a test bed for heavy duty turbocharged diesel engine. The installation (Fig. 1) includes a direct injection diesel engine, four-stroke and straight six. This engine develops maximum power of 254 kW at 2200 rpm. The engine is coupled with a water cooled eddy current brake. Measurement equipments and a rapid data acquisition system complete this installation. The turbocharger is composed of an inward-flow radial turbine equipped with directional blade distributor. The turbine is coupled to a single stage centrifugal compressor with backward inclined impeller blades. During the tests, different diesel engine loads were considered at different regimes. Indeed, for each engine operating point specified by the rotational speed and load parameters, the following variables were measured: air mass flow rate m; air pressures and temperatures at inlet and exit of the compressor P_i, T_i, P_o, T_o; turbocharger rotational speed N_{tc}; engine's rotational speed N_m; engine's load.

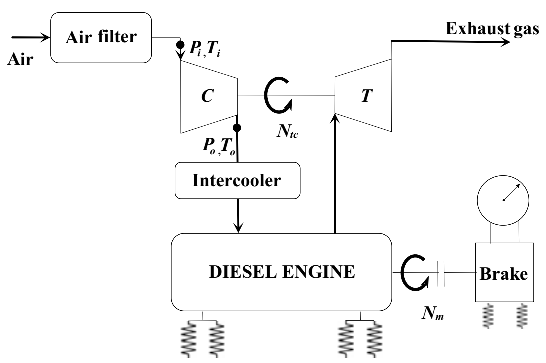


Fig. 1 Experimental setup

Five engine speeds are considered: 800, 1000, 1200, 1400 and 1800 rpm. For each speed, six loads have been considered from the low to full load. In order to make this paper concise, only the results corresponding to the average engine speed 1400 rpm are presented. The compressor efficiency and pressure ratio can be calculated using Eqs. (1) and (2).

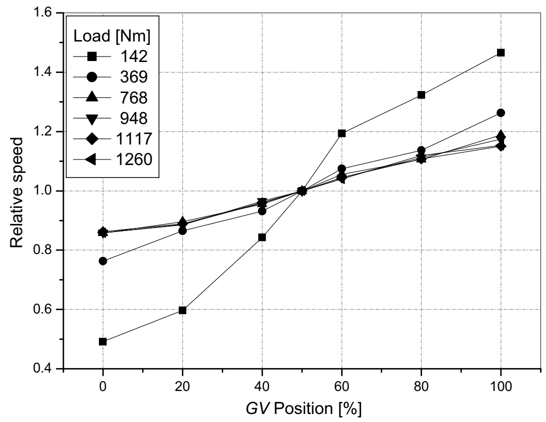
$$\eta_c = \frac{T_{o\,is} - T_i}{T_o - T_i} \tag{1}$$

where $T_{o\,is}$ is calculated as $T_{o\,is} = T_i \left(\frac{P_o}{P_i}\right)^{\frac{\gamma-1}{\gamma}}$

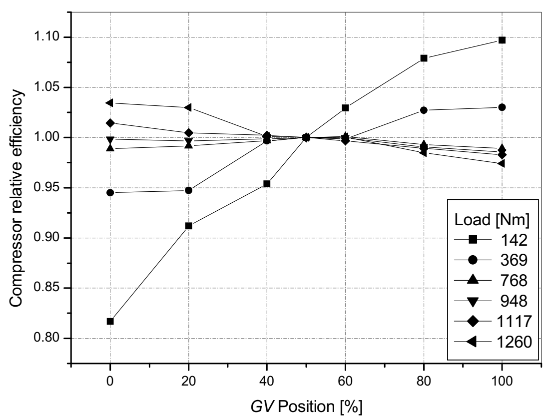
$$\pi = \frac{P_o}{P_i}\,. \tag{2}$$

3. Effect of VNT on compressor performances

To appreciate the influence of the VNT technique on compressor performances, experimental tests were performed. In fact, for each engine regime, six opening GV) positions were considered: 0%, 20%, 40%, 60%, 80% and 100%. The GV = 100% corresponds to a full opening guide vane, however the GV = 0% corresponds to the minimum distributor channel throat section.



a



b

Fig. 2 Relative performances of the compressor: a - relative speed; b - relative efficiency

Fig. 2 shows the evolution of the relative compressor performances according to GV position at different engine loads. The relative performances: rotational speed and efficiency are obtained by dividing the measured values by those corresponding to 50% GV opening (medium opening). At a given engine load, rotational speed of the compressor increases according to the GV position. This is due to a widening of the distributor channel throat section. At low engine load, the GV position influences strongly the rotational speed and efficiency of the compressor. In fact, the expansion ratio increases rapidly at low mass flow rate level corresponding to low engine load. Efficiency figure shows that the GV opening values more than 50%, lead to the decrease of efficiency for partial and full engine load. This is due to the high mass flow rate level imposed on the compressor wheel which operates under off-design conditions, Papalia et al. [6]. Japikse [7] has studied the decisive factors in advanced compressor design and development.

The off-design behaviour can be analysed by some design parameters such as fluid incidence on the wheel blades, the rotor and stator section ratio, the free space parameter, the reduced speed and position of the compressor stage on the Ns-Ds map. The inlet flow distortion leads to degradation in the compressor performances and its stable operation range, Ariga et al. [8]. An IGV design is considered to correct the fluid incidence angle at the compressor wheel inlet. The present study analyses this solution in association with the VNT to improve the compressor performance characteristics. The prewhirl technique has been substantially studied by Rodgers [9], Wallace et al. [10], Rodgers [11], Rodgers [12] and Simon et al. [13] and yet stay slight used in automotive turbocharger application, Abdullah [14]. The prewhirl offers the possibility to extend compressor's map range, Knecht [15] and is advantageous for the efficiency and surge characteristics of the compressor, Whitfield and Abdullah [16], Ishino et al. [17], Uchida et al. [18].

4. Prewhirl contribution

The prewhirl reduces the compressibility effect at convex side of the eye to avoid the formation of shock waves and the consequent losses (Fig. 3). Najjar and Akeel [5] have studied the effect of positive prewhirl on centrifugal compressor performances.

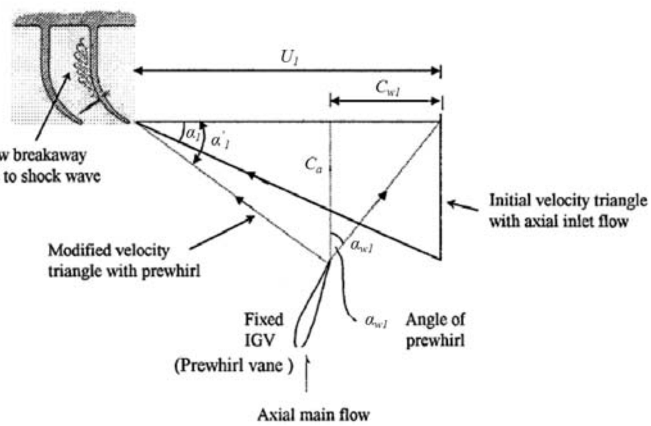


Fig. 3 Effect of prewhirl, Najjar and Akeel [5]

The authors have considered a small aircraft engine equipped with a typical compressor stage as an application. This theoretical study has shown that parabolic progression of prewhirl angle is the best. The evolution of α_{w1} angle, from 0° at root to α_{w1t} = 40° at tip radius maintains the compressor pressure ratio and work to the level of no prewhirl. Otherwise, at the wheel entry, the ratio of relative Mach numbers Mw1r defined by the equation (8) nearly decreases by half.

Losses in compressor's stage consist of viscous effects, leakage effects and aerodynamic effects. The influence of M_{w1} on these aerodynamic losses is certain. To analyze this influence on the compressor efficiency, a compressor model was performed and published, Liazid et al [19]. The approach consists to compute partial losses of the compressor stage. The assessment of these losses combined with the thermodynamic relationships describing the compression process allows the emergence of the model. In addition, the model requires knowledge of the compressor geometric data. These were carried out using a universal optical microscope with 0.005 mm axes resolution. The results are presented in Table. The model is of a low CPU cost and does not suffer from any stability problem. The evolutions of compressor's efficiency and pressure ratio according to the engine air demand at full operating load from idle to maximum engine speed are respectively represented on Fig. 4, a and b.

Prewhirl is introduced by IGV. Assuming the same mass flow in both cases (with and without prewhirl), the axial absolute velocity C_a remains nearly the same, however the relative velocity W_1 is reduced and curvature of impeller channels at inlet is reduced, i.e. α_1 increases as shown in Fig. 3. The compressor torque ($C_{w2}r_2$ -$C_{w1}r_1$) is reduced. Assuming the tangential velocity C_{w1} is constant over the eye, $C_{w1}r_1$ will increase from root to tip. Hence, work level of the compressor depends on the considered radius at the inlet section. Since M_{w1} decreases from its maximum value at the eye tip to its minimum value at the root, it is preferable to vary α_{w1} gradually, reducing it from a maximum value at the tip to zero at the root of the eye. In the case of parabolic evolution of α_{w1} from zero at root to maximum at tip, the relative velocity profile at the wheel entry is defined as

$$C_{w1}^2 = C_a tan(\alpha_{w1r}) \quad (3)$$

where α_{w1r} is the prewhirl angle at each entry radius

$$\frac{C_{w1}^2}{tan(\alpha_{w1r})} = const = \left(\frac{C_{w1}^2}{tan(\alpha_{w1})}\right)_r = \left(\frac{C_{w1}^2}{tan(\alpha_{w1})}\right)_t \quad (4)$$

take $\left(\frac{r}{r_t}\right) = r_e$ Eq. (5) is obtained

$$tan(\alpha_{w1r}) = a\,r_e + b \quad (5)$$

$$C_{w1} = \sqrt{C_a\left[a\left(r/r_t\right) - b\right]} \quad (6)$$

$$\left(C_{w1}\,r\right)_m = \int_{r_r}^{r_t} C_a\left[a\left(r/r_t\right) - b\right] r_t d\left(\frac{r}{r_t}\right) \quad (7)$$

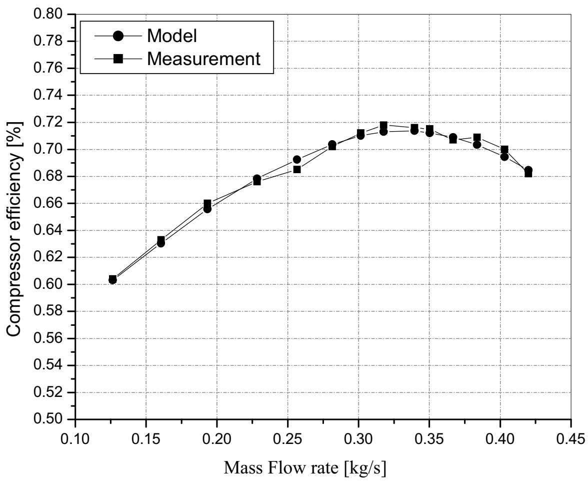
$$M_{w1r} = \frac{W_1}{\sqrt{\left(U - C_{w1}\right)^2 + C_a^2}} \quad (8)$$

Performance characteristics of the compressor are calculated by making some modifications on the computing program to take into account the prewhirl case. The same mass flow rate and compressor's rotational speed are conserved. The angle α_{w1t} was varied in the range of 10° to 40° to verify if the compressor work conservation (with and without prewhirl) is the best at α_{w1t} = 40° as found in the case of the compressor stage studied by Najjar and Akeel [5]. This calculation was conducted for all GV position range (GV = 0% to GV = 100%). Each GV position accepts a wide range of diesel engine operation (several speeds and several loads). Only the nominal regime

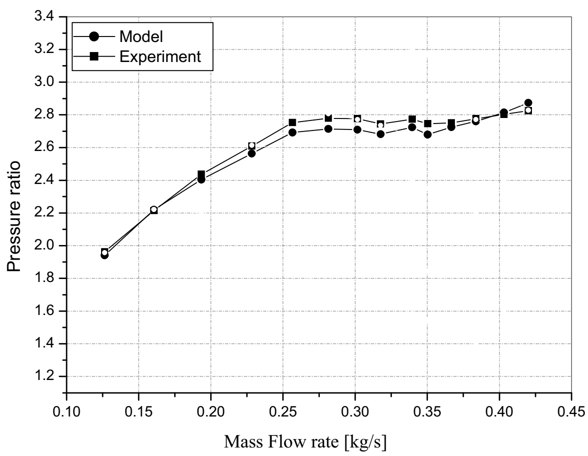
Table

Compressor's geometric data

Components	Value
Number of impeller blades Z	12
Diameter at the entry of the inlet channel D_0, mm	120
Diameter at the exit of the inlet channel D_1, mm	61
Tip diameter at the impeller inlet D_{1ext}, mm	60
Diameter at the impeller exit D_2, mm	83.5
Root diameter at the impeller inlet D_{ib}, mm	19
Impeller hydraulic length L_R, mm	46.5
Diffuser exit diameter D_3, mm	117
Volute exit section A_4, m²	2.55E-03
Volute exit radius r_4, mm	60
Mean blade angle at the impeller inlet β_{1b}, °	150
Blade angle at the impeller exit β_{2b}, °	126



a



b

Fig. 4 Prediction capability of the compressor model: a - compressor efficiency; b - pressure ratio

N = 1400 rpm is considered under six different loads from low to thigh values. The results are presented in the ratio form. The reference is the performance of the compressor without prewhirl. The obtained results indicate that for our application the best value is α_{w1t} = 10°. This is because our compressor is 6 times smaller than that studied by Najjar and Akeel [5]. Figs. 5 and 6 give respectively an example of the compressor relative work evolution and relative pressure ratio evolution at partial engine load according to the GV position for all α_{w1t} range.

Hence using α_{w1t}=10° and r_e=0.316, the Eq (6) becomes

$$C_{w1} = \sqrt{C_a\left[1.0622\left(r / r_t\right) - 0.335\right]} \tag{9}$$

and

$$\left(C_{w1}\, r\right)_m = \int_{r_r}^{r_t} C_a\left[1.0622\left(r / r_t\right) - 0.335\right] r_t\, d\left(\frac{r}{r_t}\right) \tag{10}$$

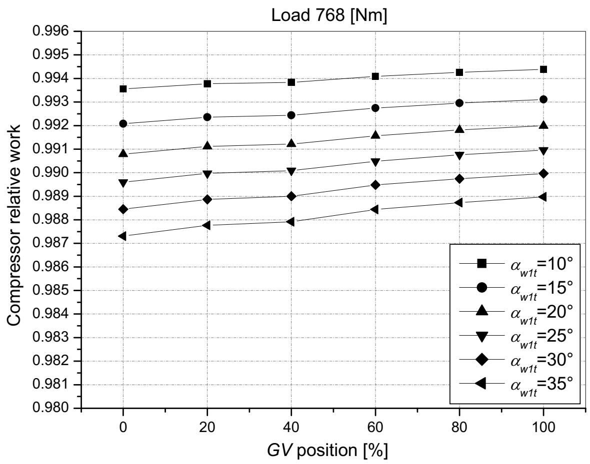


Fig. 5 Compressor relative work

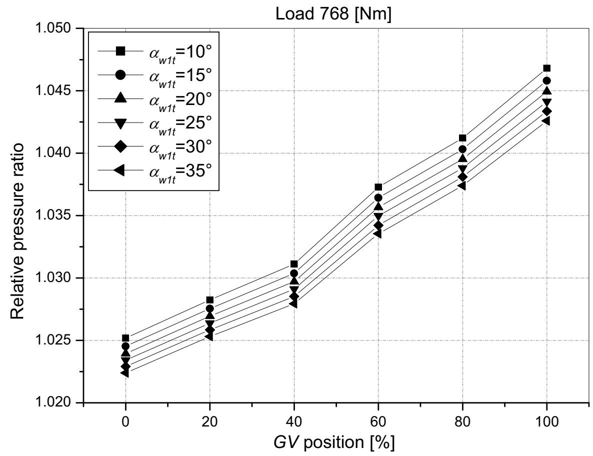


Fig. 6 Relative pressure ratio of the compressor

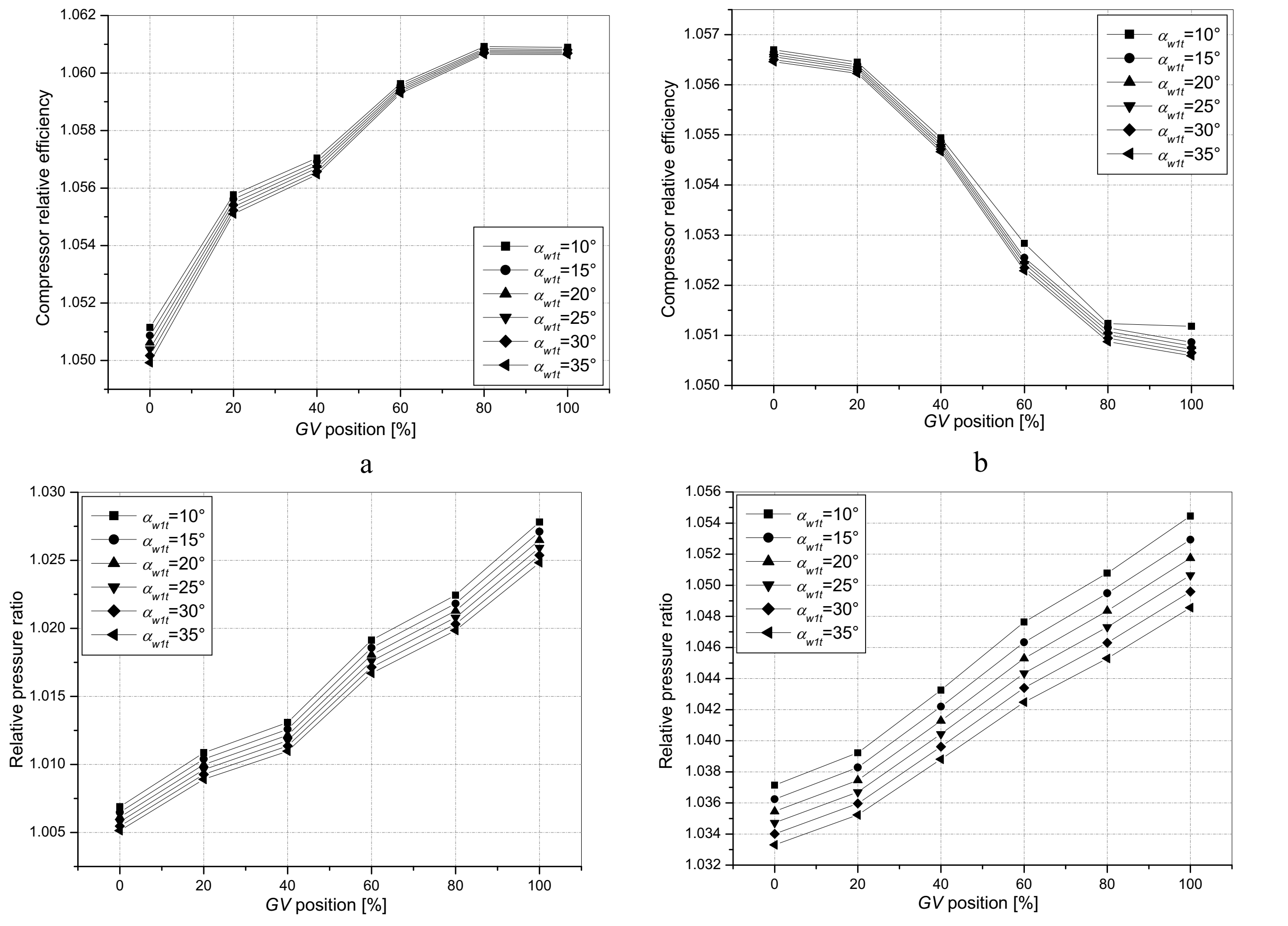


Fig. 7 Relative performances of the compressor: a - relative efficiency for 369 Nm load; b - relative efficiency for 1117 Nm load; c - relative pressure ratio for 369 Nm load; d - relative pressure ratio for 1117 Nm load

Fig. 7 illustrates examples of the compressor's performance characteristics. It should be noted that the prewhirl improves efficiency without adjusting the general look of the efficiency curve at high engine loads. The explanation can be provided by the analysis of losses in the compressor's stage according to the GV position (Fig. 8). The predominant are the power losses and diffuser losses. It is clear that the tendency of compressor's efficiency is mainly governed by these two types of losses. The reduction of these ones can be achieved by using a Guide Vanes Diffuser (GVD) and an adaptation of the variable blade angle at the compressor wheel exit. So this solution is actually technically difficult to design. Fig. 9 shows the reduction ratio of incidence losses by prewhirl over all GV range and all engine loads. It can be noticed that the incidence loss reduction remains constant with the full GV opening over all the engine load range. However, this reduction increases as the GV closes and the load decreases. This indicates that the prewhirl effect is better for low air flow rates at the compressor inlet because the flow angle is best established than the case of high flow rates.

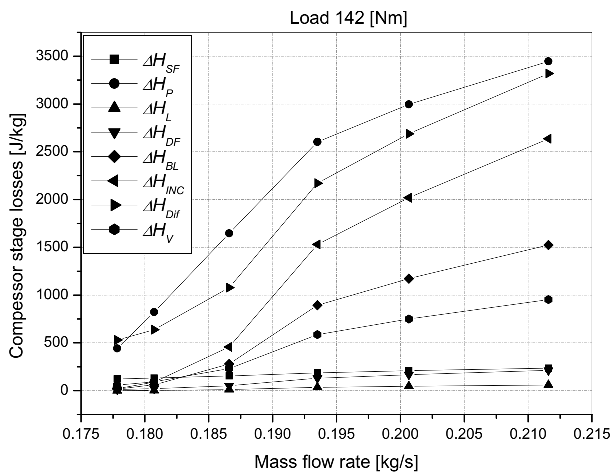


Fig. 8 Compressor stage losses

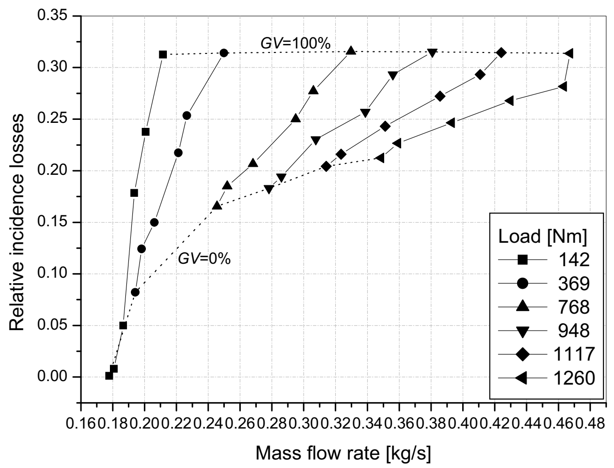


Fig. 9 Incidence losses reduction by prewhirl

5. Conclusion

This paper concerns the effects of VNT and IGV association on Turbocharger Centrifugal Compressor Performances. The study shows that for the high engine loads, a drop in compressor's efficiency is observed notably because of the shock effect.

A prewhirl design at the compressor inlet is investigated as a solution. It appears that the positive prewhirl designed as a parabolic profile along the radius at the inlet section of the compressor wheel improves its performance. The synchronization between the position of GV and the maximum angle of prewhirl at the compressor's inlet was made under the constraint of highest level of compressor work and pressure ratio conservation. This study shows the favourable impact of the prewhirl on performances of the compressor but without adjusting the look of the efficiency curve at large GV openings. A future work remains to be developed to reduce the level of losses detachment at the wheel exit as well as corresponding diffuser losses.

References

1. **McCutcheon, A.R.S.; Brown, M.W.G.** 1986. Evaluation of a variable geometry turbocharger turbine on a commercial diesel engine. IMechE Conference Publications, Paper n. C104/86: 179-189.
2. **Capobianco, M.; Gambarotta, A.** 1992Variable geometry and wastegate automotive turbochargers: measurements and comparison of turbine performance, Journal of Engineering for Gas Turbines and Power, vol.114, n.3: 553-560.
3. **Okazaki, Y.; Matsudaira, N.; Hishikawa, A.** 1986. A case of variable geometry turbocharger development. -IMechE Conference Publications, paper n. C111/86: 191-195.
4. **Binder, N.; Carbonneau, X.; Chassaing, P.** 2005. Influence of a variable guide vane nozzle on the design parameters of a radial turbine stage. Proc of the 6th European Conference on Turbomachinery, Lille, France.
5. **Najjar, Y.S.H.; Akeel, S.A.M.S.** 2002. Effect of prewhirl on the performance of centrifugal compressors. -International Journal of Rotating Machinery, vol.8, issue 6: 397-401.
6. **Papalia, J.; Lawless, P. B.; Fleeter, S.** 2005. Off-design transonic rotor-inlet guide vane unsteady aerodynamic interactions, Journal of Propulsion and Power, vol.21, n.4: 715-727.
7. **Japikse, D.** 2000. Decisive Factors in Advanced Centrifugal Compressor Design and Development Concepts. ETI, Inc.
8. **Ariga, I.; Masuda, S.; Ookita, A.** 1987. Inducer stall in a centrifugal compressor with inlet distortion, Trans. ASME, Journal of Turbomachinery, vol.109: 27-35.
9. **Rodgers, C.** 1964. Typical performance of gas turbine radial compressors, Trans. ASME, Series A, vol.86: 161-175.
10. **Wallace, F.J.; Whitfield, A.; Atkey, R.C.** 1975. Experimental and theoretical performance of a radial flow turbocharger compressor with inlet prewhirl. Proc. IMechE, vol.189: 43-75.
11. **Rodgers, C.** 1977. Impeller stalling as influenced by diffusion limitations, Trans. ASME, Journal of Fluids Engineering, vol.99: 84-97.
12. **Rodgers, C.** 1978. A diffusion factor correlation for centrifugal impeller stalling, Trans. ASME, Journal of Fluids Engineering, ASME Paper n. 78-GT-61.
13. **Simon, H.; Wallmann, T.; Mönk, T.** 1987. Improvements in performance characteristics of single-stage and multistage centrifugal compressors by simultaneous adjustments of inlet guide vanes and diffuser vanes,

Trans. ASME Journal of Turbomachinery, vol.109: 41-47.
14. **Abdullah, A.H.** 1996. The Application of High Inlet Swirl Angles for Broad Operating Range Turbocharger Compressor. Ph.D. dissertation, Bath Univ.; England, UK.
15. **Knecht, W.** 2006. Diesel development in view of reduced emission standards. Proc. 19th International Conference on Efficiency, Cost, Optimization, Simulation, and Environmental Impact of Energy Systems ECOS 2006, Aghia Pelagia, Crete, Greece, vol.2: 8-16.
16. **Whitfield, A.; Abdullah, A.H.** 1998. The performance of a centrifugal compressor with high inlet prewhirl, Trans. ASME Journal of Turbomachinery, vol.120, n.3: 487-493.
17. **Ishino, M.; Iwakiri, Y.; Bessho, A.; Ushida, H.** 1999. Effect of variable inlet guide vanes on small centrifugal compressor performance, ASME Paper, n.99-GT-157.
18. **Uchida, H.; Kashimoto, A.; Iwakiri, Y.** 2006, Development of wide range compressor with variable inlet guide vane, R&D Review of Toyota, vol.41, n.3: 9-14.
19. **Liazid, A.; Izidi, L.; Bencherif, M.** 2004. Prédiction des performances d'un compresseur centrifuge de suralimentation, Revue Sciences & Technologie B, Constantine, Juin 2004, N°21: 15-21.

L. Izidi, A. Liazid

REGULIUOJAMŲ TŪTOS IR ĮĖJIMO KREIPIAMŲJŲ MENČIŲ ĮTAKA PRIPŪTIMO IŠCENTRINIO TURBOKOMPRESORIAUS CHARAKTERISTIKOMS

R e z i u m ė

Vidaus degimo variklių naudingumo koeficientui pagerinti plačiai naudojami turbininiai pripūtimo kompresoriai. Kompresoriaus charakteristikos turi didelę reikšmę, nes nuo jų priklauso į variklio cilindrus tiekiamo oro kiekis, bei oro ir kuro santykis, nuo kurio priklauso degimo kokybė. Apskritai geras variklio ir turbininio pripūtimo suderinamumas pasiekiamas keičiant jo konstrukciją. Šiame straipsnyje nagrinėjama turbinos tūtos ir įėjimo kreipiamųjų derinimo įtaka kompresoriaus charakteristikoms. Deja, išskyrus pramoninius kompresorius, srauto kryptis automobilių kompresorių įėjime nereguliuojama, nors jų sukimosi greitis didelis. Atliekant šį darbą eksperimentinis tyrimas buvo derinamas su skaitmenine analize. Tobulinant įėjimo kreipiamąsias mentes pavyko realiai pagerinti kompresoriaus charakteristikas visame jo darbo diapazone, tačiau nekreipiamas dėmesys į efektyvumo kreivę, kai turbinos mentės plačiai atidarytos ir patiriama didelių galios ir difuzijos nuostolių.

L. Izidi, A. Liazid

EFFECTS OF VNT AND IGV ASSOCIATION ON TURBOCHARGER CENTRIFUGAL COMPRESSOR PERFORMANCES

S u m m a r y

Turbochargers are today widely used to improve performances of internal combustion engines. The compressor's characteristics are important because they govern the fresh air quantity introduced to engine cylinder and consequently determine the air/fuel ratio which governs the combustion quality. Generally, the best engine-turbocharger matching is achieved by using the variable geometry technique. This paper studies the effects of VNT (Variable Nozzle Turbine) and IGV (Inlet Guide Vanes) combination on compressor's characteristics. However, except the industrial compressor cases, the flow direction adjustment at the entry of automotive compressor is not yet applied in spite of the high rotational speed. An experimental investigation coupled with a numerical study is used to achieve this work. The IGV technique shows a real improvement of the compressor's characteristics over its whole operating range but without adjusting the look of the efficiency curve at large opening of turbine nozzle blades because power and diffuser loses remain important.

Л. Изиди, А. Лязид

ВЛИЯНИЕ РЕГУЛИРУЕМЫХ СОПЛА И ЛОПАТОК ВХОДНЫХ НАПРАВЛЯЮЩИХ НА ХАРАКТЕРИСТИКИ НАДДУВА ЦЕНТРИФУГНОГО ТУРБОКОМПРЕССОРА

Р е з ю м е

Турбокомпрессоры в настоящее время широко используются для повышения коэффициента полезного действия двигателей внутреннего сгорания. От характеристик компрессора зависит количество подаваемого в цилиндры двигателя воздуха и таким образом, устанавливается соотношение воздух/горючее, определяющее качество сгорания. Хорошее согласование пускового компрессора и двигателя можно достичь изменением конструкции компрессора. Эта статья рассматривает влияние наладки сопла турбины и входных направляющих на характеристики компрессора. К сожалению, применяемое в промышленных компрессорах регулирование направления потока в автомобильных компрессорах не используется, несмотря на высокую скорость вращения. При проведении настоящей работы использовано экспериментальное исследование совместно с числовым анализом. Результаты совершенствования лопаток входных направляющих показывают реальное улучшение характеристик компрессора во всем диапазоне рабочих режимов, если необращать внимания на кривую эффективности при большом открытии лопаток сопла турбины, так как потери мощности и диффузии в этом случае остаются значительными.

Received August 02, 2010
Accepted April 05, 2011