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Architecture Study & Performance Analysis Of E-Compressor in a Naturally Aspirated Gasoline Engine

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Abstract : Compressors play a vital role in vehicle air conditioning systems that mainly impact vehicle performance and occupant comfort. As automotive technologies evolve to meet the demands of sustainability and efficiency, the integration of electrified components into internal combustion engine (ICE) vehicles has become a focal point of research and development. Conventionally, ICE vehicles use belt-driven compressors, which use engine power from the auxiliary belt drive to operate. In this context, our study delves into the integration of an electric compressor (e-compressor) within the Entry segment vehicle platform, offering a detailed examination encompassing an architecture layout study, performance analysis, and cost implications.

Keywords: E-compressor, Belt-driven compressor, Cooling capacity, Layout study, Coefficient of performance

Introduction

Compressors are integrated in an internal combustion engine vehicle (ICE) for the application of air conditioning systems that significantly impact the vehicle performance and occupant comfort. Conventionally, belt-driven compressors have been integrated with ICE vehicles to meet the passenger cooling requirements. With the emergence of electric and hybrid vehicles, e-compressors have opened new opportunities for improved vehicle performance [1] and efficiency [2] by increased fuel economy[3], cabin cool down period and lesser emissions[4]. Vapor compression (Fig.1)[5] plays a very important role when it comes to compressors. Vapour compression works on the principle of transferring heat from a cold reservoir to a hot one. This is against the second law of thermodynamics which makes it necessary for some work to be done. The vapor compression refrigeration cycle involves various components namely the compressor, condenser, heat exchanger, evaporator, and the expansion valve/throttle valve.

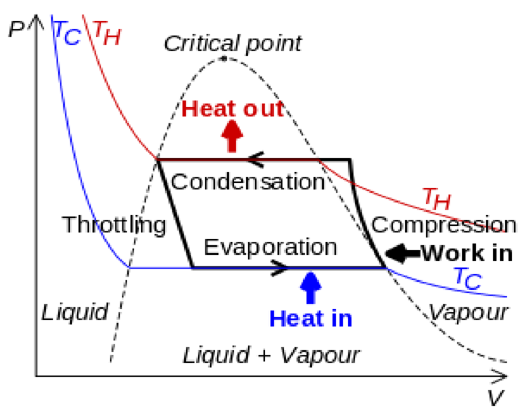


Fig.1 Vapor compression cycle[5]

For every HVAC unit a refrigerant is used as cooling medium. R134a refrigerant, composed of tetrafluoroethane, is commonly used in automotive air conditioning systems in India. As a hydrofluorocarbon (HFC), R134a is recognized for its efficiency in cooling and has become a popular replacement for ozone-depleting chlorofluorocarbon (CFC) refrigerants like R12. Known for being ozone-friendly and free from chlorine, R134a contributes to environmental sustainability. While R134a remains a prevalent choice, newer refrigerants like HFO-1234yf are being introduced in some regions to further minimize environmental impact. In India, the commercialized refrigerant that is used in most of the passenger vehicles, R134a is taken into consideration for our study. The chemical and physical properties of R134a refrigerant is given in Table 1.

Table 1. Properties of refrigerant R134a

SNO.	PROPERTIES	VALUE
1	Chemical formula	CH_2FCF_3
2	Molar mass	102.03 kg/kmol
3	Critical pressure	4.06 MPa
4	Maximum pressure	70 MPa
5	Critical point density	511.9 kg/kmol
6	Critical temperature	374 K
7	Boiling point temperature	247.08 K
8	Specific heat @ const.P(S.L @ 25°c)	1.4246
9	Specific heat @ const.P (S.V @ 25°c)	1.0316

There are mainly two types of automotive air conditioners: one is a Thermal Expansion Valve (TXV) system and the other is a Clutch Cycling Orifice Tube (CCOT) system[6]. The TXV system regulates rate of refrigerant flow into evaporator as governed by evaporator outlet pipe temperatures sensed by the sensing bulb, and the CCOT system controls the evaporator temperature by turning the compressor on and off with a clutch cycling switch. A CCOT automotive A/C system includes a compressor, a condenser, an expansion device, an evaporator and an accumulator. The system taken into consideration is a TXV A/C system.

In a TXV A/C system the expansion valve performs two main functions namely, maintain the pressure differential between low- and high-pressure sides and control the amount of liquid refrigerant entering the evaporator. The final stage of refrigerant compression cycle is evaporation where the refrigerant is at a lower temperature than its surroundings. So, when it evaporates and absorbs the latent heat of vaporization. This extraction of heat from refrigerant takes place at low pressure and temperature. The compressor's suction helps maintain the low pressure.

One major component is the air-cooled evaporator which removes the humidity from the cabin air and makes it dry. Upon entering the evaporator, the refrigerant is in a cold, partial liquid-vapor mixture. The evaporator's operating pressure and temperature, known as suction pressure and suction temperature, respectively, influence its performance. The suction line acts as the conduit for transporting refrigerant gas from the evaporator to the compressor. The primary function of the evaporator is to transition the refrigerant from a partial liquid-vapor state to a fully saturated vapor. The working of an air-cooled evaporator is shown in Fig.2.

Throughout this process, heat is transferred to the refrigerant, maintaining a constant temperature as the remaining liquid converts to gas, resulting in a 100% vapor saturation output at the initial constant temperature. Any additional heat introduced to the 100% vapor refrigerant leads to an increase in temperature, which is defined as superheat. [7]

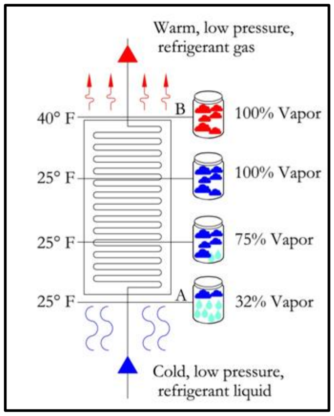


Fig.2 Working of evaporator [6]

Other major component in the HVAC loop is the air-cooled condenser unit which is a heat exchanger that removes heat from refrigerant vapor. The refrigerant, having transitioned from the compressor, now exists as a hot, high-pressure gas upon entering the condenser. This superheated gas undergoes a cooling process, gradually reaching its saturation temperature. Heat dissipation occurs as the hot refrigerant gas releases heat to the condenser's cooling medium. Subsequently, the 100% saturated vapor transforms into 100% saturated liquid. During this phase, heat is expelled to the condenser cooling medium, causing the vapor to condense into a liquid. Finally, the 100% saturated liquid undergoes sub-cooling. In an ideal condenser, sub-cooling does not occur, and once the refrigerant achieves a fully saturated liquid state, any additional heat loss results in a reduction in temperature. This cooling of the saturated liquid is termed sub-cooling.[7]

This study deeply dwelves into the selections of compressors, the architecture layout study, modifications required for the adaptation, detailed part level performance analysis and the cost analysis of the replacement of conventional belt-driven compressor by e-compressor.

Selection of Components

From Fig.3 the schematic layout of adapting an e-compressor in an ICE vehicle can be seen. The e-compressor is mounted near the powertrain compartment on the vehicle body.

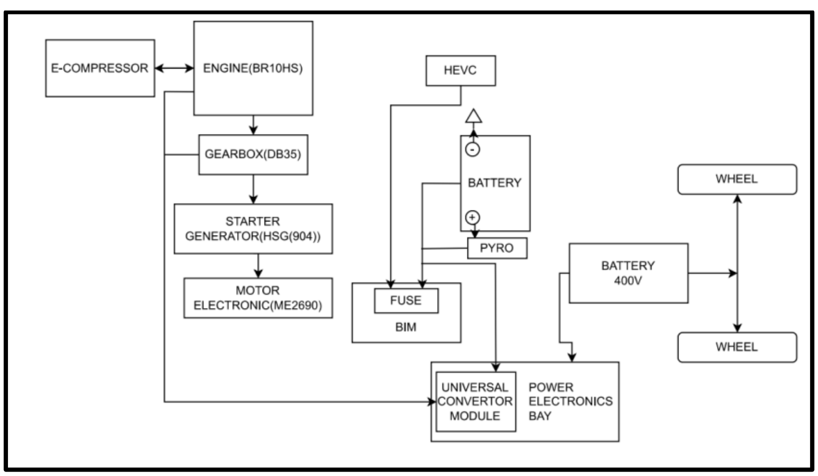


Fig.3 Schematic layout of e-compressor integration in an ICE vehicle

Despite the drawbacks in cabin temperature distribution and cooling effect observed in the DC compressor for automobile AC systems, its overall performance surpasses that of conventional compressors. The electric compressor demonstrates superior efficiency and effectiveness compared to its traditional counterparts, despite the noted limitations in cabin temperature distribution and cooling[8][9]. Below shown are the selection of various components for the replacement of conventional compressor.

I. Selection of Compressors:

ICE vehicles use belt-driven compressors for their simplicity in design and easy maintenance. Belt-driven compressors need no additional power source i.e., it can be powered by the engine itself. Also, belt-driven compressor can provide better cooling performance at high engine speeds making them effective to use in hot weather or demanding driving conditions.

For the conventional belt driven compressor, piston type, variable displacement compressor was used for the study as shown in fig.4. The technical specifications are as mentioned in Table 2. The belt-driven compressor in consideration is designed with a 29.3mm bore and variable stroke ranging from 1.9mm to 30.9mm, showcasing adaptability to diverse operating conditions. As a piston-type compressor with variable displacement, it efficiently compresses R-134a refrigerant, a widely used and environmentally friendly cooling agent. The displacement capacity spans from 7.5 cm^3 to 125.11 cm^3 per revolution, indicating versatility in handling different refrigeration requirements. The compressor's resilience is evident in its burst pressure specifications, with the high-pressure side capable of withstanding up to 12.4 MPa and the low-pressure side up to 4.81 MPa. Overall, this compressor stands out for its adaptability, efficiency, and robust design, making it suitable for various air conditioning or refrigeration applications.

Table 2. Specifications of belt-driven compressor

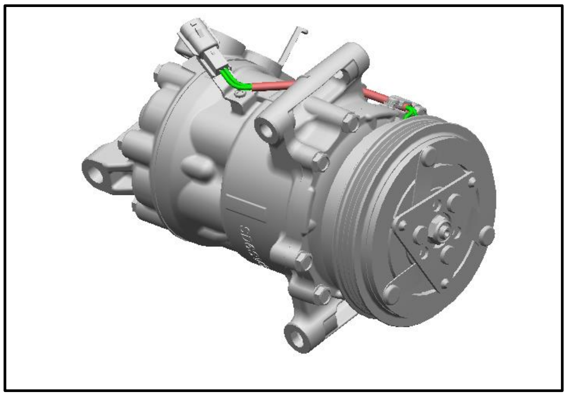


Fig.4 Conventional belt-driven compressor

The e-compressor in consideration as shown in fig 5. has a capacity of 27cc, scroll type operating within a versatile speed range of 1000 to 3200 rpm. With a working temperature range spanning from -10°C to 85°C, this e-compressor is designed to function efficiently across a spectrum of environmental conditions. It utilizes POE68 oil for lubrication, ensuring smooth operation and longevity. The high-voltage (HV) requirement falls between 40-60VDC, and despite its powerful performance, the e-compressor maintains a relatively lightweight profile at 6.4kg.

Notably, it maintains a suction pressure of 0.3MPa and a discharge pressure of 1.5MPa, indicative of its capability to handle varying pressures in different operational scenarios. The degree of superheating and supercooling is specified at 10K and 5K, respectively, highlighting the precision and control embedded in this e-compressor's design. The technical specifications of the e-compressor is given in Table 3. Overall, these specifications underscore its adaptability, efficiency, and reliability for diverse applications within a range of temperature conditions.

S.NO	CONTENTS	PARAMETERES
1	Bore	29.3mm
2	Stroke(min-max)	1.9-30.9mm
3	Type	Piston type – Variable displacement
3	Displacement (min-max)	7.5-125.11 cm^3//rev
4	Refrigerant	R-134a
5	Burst pressure	HP side: 12.4 MPa LP side: 4..81 MPa

Table 3. Specifications of e-compressor

S.NO	CONTENTS	PARAMETERES
1	Capacity	27cc
2	Speed range	1000 – 3200 rpm
3	Working temperature range	-10°c to 85°c
4	Oil	POE68
5	HV Voltage	40-60VDC
6	Weight	6.4kg
7	Suction pressure	0.3MPa
8	Discharge pressure	1.5MPa
9	Degree of superheating	10K
10	Degree of supercooling	5K

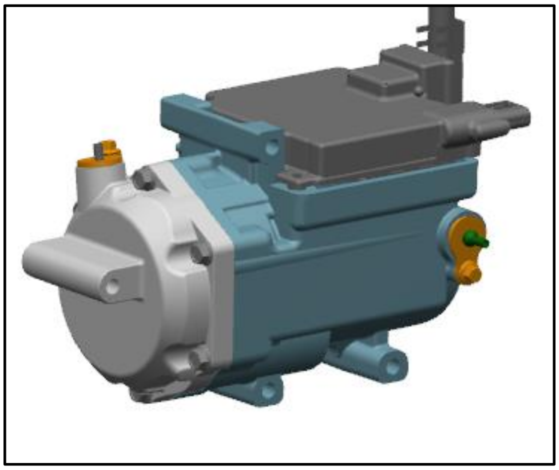


Fig.5 Scroll Type e - compressor

The Table 4. outlines the cooling capacity performance of the specified e-compressor across different rotational speeds. At 1000 rpm, the e-compressor demonstrates a cooling capacity of 0.84 kW with a power consumption of 0.47 kW, resulting in a Coefficient of Performance (COP) of 1.78.

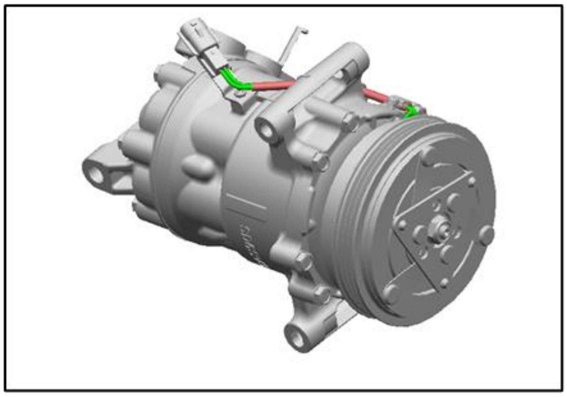
As the rotational speed increases to 2000 rpm, the cooling capacity nearly doubles to 1.70 kW, accompanied by a power input of 0.77 kW, yielding a higher COP of 2.23. Continuing the trend, at 3000 rpm, the e-compressor achieves a cooling capacity of 2.63 kW with a power consumption of 1.16 kW, maintaining a relatively efficient COP of 2.26.

These results explains the e-compressor's ability to deliver higher cooling outputs as the rotational speed increases, showcasing its adaptability and efficiency in meeting varying cooling demands. The COP values indicate the energy efficiency of the e-compressor, with higher values representing more effective cooling performance per unit of power consumed.

Table 4. Cooling capacity of e-compressor

Speed / rpm	Cooling capacity / kW	Power/ kW	COP
1000	0.84	0.47	1.78
2000	1.70	0.77	2.23
3000	2.63	1.16	2.26

Fig.6 shows the structural comparison between the conventional belt driven compressor and e-compressor taken into consideration.



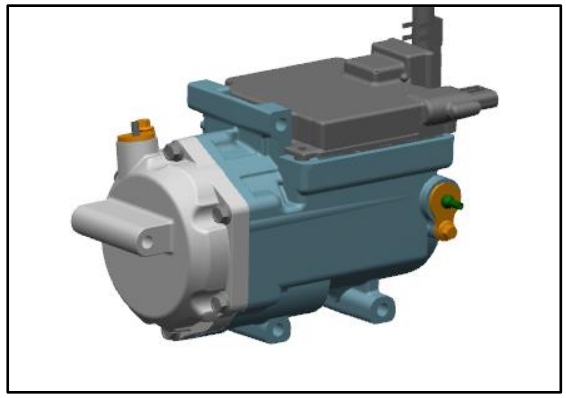


Fig.6 Depicts the structural comparison between belt-driven compressor and e-compressor.

Architecture Integration study

I. Architecture Layout Study:

The integration of electrified components is pivotal in optimizing vehicle architecture. In our analysis, the packaging of both conventional and scroll-type e-compressors within the Entry Segment Platform is meticulously examined. By comparing these configurations, we aim to uncover insights into spatial requirements, potential design challenges, and the overall impact on the vehicle's layout. Special attention is given to the distinctive characteristics of e-compressors, exploring how their compact design and smooth operation influence the integration process.

In the examination of the architectural layout within the Entry Segment Platform vehicle platform, our study encompassed both belt-driven and electric compressors, revealing a strategic placement of both components within the front zone of the engine bay area. From the Fig.7 we can see that the belt-driven compressor is mounted behind the cooling module of the vehicle. Several criteria were considered during the packaging process of the belt-driven compressor in this specific zone. The assessment involves examining potential water splash exposure from the wheel well to the engine bay area, ensuring that the compressor remains protected from such elements.

Additionally, careful consideration is given to the positioning and layout of cooling hoses connected to the compressor, a crucial aspect of the packaging process. The belt-driven compressor follows an assembly procedure where the belt is pre-assembled in a slack manner with the powertrain unit before the initiation of the docking sequence. The integration of these components is particularly significant as it occurs before the docking of the powertrain unit. In the case of the belt-driven compressor, special attention is directed toward the oil drain area to enhance overall packaging efficiency. These factors collectively contribute to the architectural packaging of the belt-driven compressor, ensuring its resilience and optimal functionality in the intended operational environment.

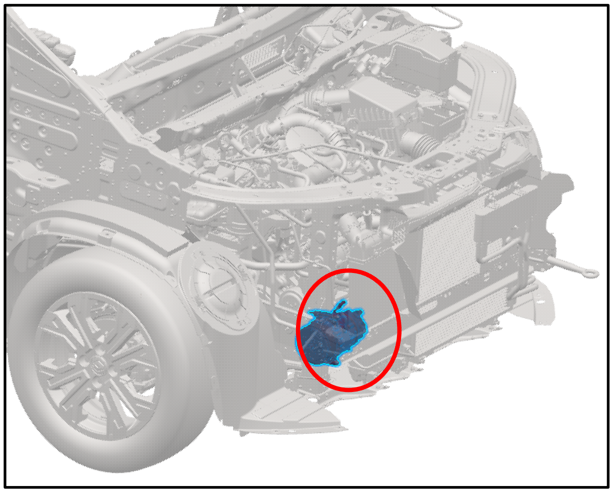


Fig.7 Isometric view of belt-driven compressor in entry segment vehicle platform

Fig.8 represents the mounting of e-compressor in the entry segment platform vehicle . A key aspect of the layout optimization involved considerations for tool accessibility during the assembly process. The e-compressor was assembled on the body of the vehicle, enhancing tool accessibility and facilitating an efficient assembly process. This e-compressor is mounted on the body with the help of stiffened rubber mounting to eliminate the NVH caused by the compressor in load conditions. Also, the e-compressor is packaged in a way that the HV harness from the DC-DC invertor is connected to the compressor.

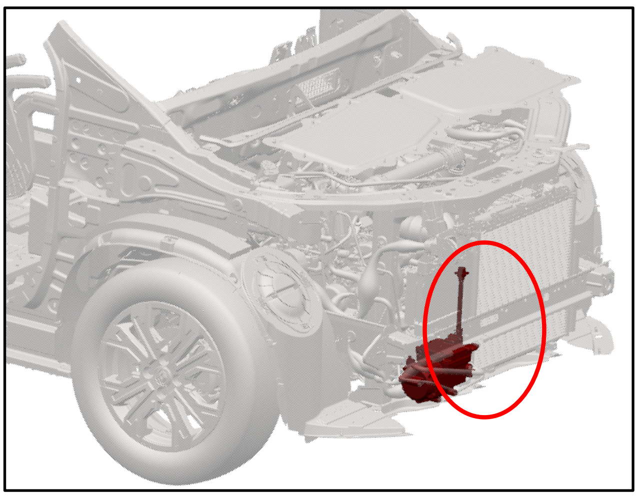


Fig.8 Isometric view of e-compressor in Entry segment vehicle platform

This thoughtful packaging layout, tailored to the specific needs of each compressor type, not only ensures efficient assembly during the manufacturing process but also contributes to the overall functionality and accessibility of the components within the Entry Segment Platform vehicle platform. By placing the compressors in relation to the powertrain unit and incorporating a tailored assembly approach, the architectural layout enhances both the manufacturing efficiency and the operational performance of the integrated compressor systems.

II. Packaging Impacts:

Packaging study was carried out with replacement of belt driven compressor with e-compressor in Entry Segment Vehicle platform, which shows the impact of integrating an e-compressor with ICE powertrain. The integration of an e-compressor into an internal combustion engine (ICE) vehicle primarily influences body interfaces, hose routing, and harness routing. The attachment of the compressor to the Body in White (BIW) necessitates the precise definition of interfaces, particularly in terms of cooling hose configuration. The adaptation process involves ensuring integration of the electrical components with the vehicle structure and optimal routing of hoses and harnesses. These considerations highlight the key impact areas associated with the adaptation of an e-compressor, emphasizing the importance of planning for implementation within the ICE vehicle architecture. Other Impacts on Packaging Includes , the need to improve the NVH, Cooling Pipes & Interface Brackets Contribute to the Packaging Environment.

III.Modifications required:

When integrating an e-compressor in entry-segment ICE vehicle there need to be addition of several electronic components to the vehicle.

i. DC-DC Invertor:

Electric compressors typically require a specific voltage level to operate optimally. In an electric vehicle, the main power source is the high-voltage battery, which may operate at a voltage level different from what the e-compressor needs. The DC-DC inverter as shown in Fig.9 facilitates the conversion of the high-voltage DC power from the EV's battery to the voltage level required by the e-compressor, ensuring compatibility and efficient operation. By addition of DC-DC invertor the harness routing of the electrical components are also modified accordingly.

Fig.9 DC-DC Invertor

ii. Body Interfaces:

To enhance the assembly feasibility of the e-compressor, necessary modifications to the body are imperative. Simultaneously, adaptations to the BIW are also mandated to accommodate the unique packaging requirements of the e-compressor. For mouting the compressor on BIW specifically stiffened rubber mounts are employed for better NVH(Noise, Vibration & Harshness) characteristics of both the compressor and the body parts. These adjustments are crucial for ensuring seamless integration and optimal performance of the e-compressor within the overall vehicle architecture.

iii. BSG:

As shown in Fig.10 the belt-driven starter generator in automobiles plays a dual role: initiating engine startup by turning the crankshaft during ignition and, once the engine is operational, serving as a generator to transform mechanical energy into electrical power for the vehicle's systems. This is facilitated by a robust belt connecting the starter generator to the engine's crankshaft, ensuring the efficient transfer of mechanical energy. Regarding its integration with a DC-DC inverter, the starter generator generates AC power, which is subsequently converted to DC to supply electrical power to various car components. An additional noteworthy feature is its potential contribution to energy efficiency through regenerative braking, where the system captures and converts braking energy into electrical energy, further enhancing the overall efficiency of the vehicle.

Fig.10 Belt driven starter generator

Performance Analysis

Both the belt driven compressor and e-compressor were tested under two test conditions as shown in tables 5, 6, 7 & 8. The nomenclature used are as follows,

Pd: Discharge pressure

Ps: Suction pressure

Nc: Speed of compressor in RP,

Q: Cooling capacity in Kw

L: Power in kW

COP: Coefficient of Performance

S/H: Superheat

S/C: Supercooled

Performance Analysis of belt-driven compressor:

In test condition 1 as shown in Table 5 the Pd/Ps ratio(Pressure Ratio) is taken as 1.5/0.3. The performance evaluation of a belt-driven compressor under specified temperature differentials, denoted as S/H 10K and S/C 5K, with varying compressor speeds are discussed. From the table.5 it is evident that, as the rotational speed (Nc) of the compressor increases from 2000 RPM to 6000 RPM, the power consumption also rises from 1.77KW to 5.59KW(Fig.10). This suggests that higher rotational speeds demand more electrical power for the compressor operation.

Table 5. Performance test results of test condition 1 with Pd/Ps = 1.5/0.3

Condition	Pd/Ps (Mpa)	Nc(RPM)	Cooling Capacity Q (KW)	Power L (kW)	COP	Discharge T(°c)
S/H 10K S/C 5K	1.5/0.3	2000	2.7	1.77	1.525	<120
		4000	4.49	3.17	1.416	<120
		6000	7.9	5.59	1.307	<120

One notable inference from the provided data is that increasing the compressor speed has a significant impact on the refrigeration system's performance. As the compressor speed rises from 2000 to 6000 RPM, there is a consistent trend of higher cooling capacity(Fig 10), indicating an enhanced ability to remove heat from the system. However, this comes at the expense of increased power consumption(Fig 10), as reflected in the escalating values for power consumption at higher speeds. Despite the higher power consumption, the system's efficiency, measured by the coefficient of performance (COP), experiences a slight decline with increasing compressor speed(Fig 11). This suggests a trade-off between achieving greater cooling capacity and maintaining optimal energy efficiency. Also the discharge temperature is effectively controlled and remains below 120°C across all tested conditions, emphasizing the system's ability to operate safely within specified temperature limits. This inference underscores the importance of carefully balancing compressor speed to meet performance requirements while considering the associated energy consumption and system efficiency trade-offs.

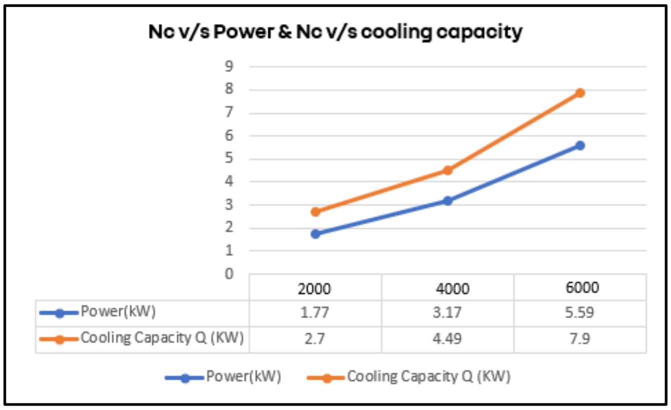


Fig.10 Nc v/s Power(kW) & Nc v/s Cooling capacity for test condition 1

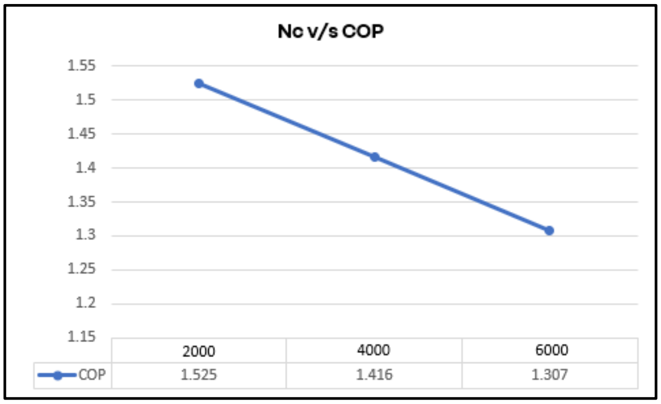


Fig.11 Nc v/s COP for test condition 1

In test condition 2 as shown in table 6 the Pd/Ps ratio(Pressure Ratio) is taken as 2.1/0.4. The performance evaluation of a belt-driven compressor under specified temperature differentials, denoted as S/H 10K and S/C 5K, with varying compressor speeds are discussed. The pressure ratios of 2.1/0.4 delineate the compression characteristics. The escalation in compressor speed from 2000 to 6000 RPM corresponds to a significant increase in cooling capacity, peaking at 13.6 kW at the highest speed(Table 6).

Table 6. Performance test results of test condition 2 with Pd/Ps = 2.1/0.4

Condition	Pd/Ps (Mpa)	Nc(RPM)	Cooling Capacity Q (KW)	Power L (kW)	COP	Discharge T(°c)
S/H 10K S/C 5K	2.1/0.4	2000	4.66	2.67	1.74	<120
		4000	7.73	5.05	1.53	<120
		6000	13.6	8.911	1.32	<120

However, this enhanced performance is counterbalanced by an elevation in power consumption, surging from 2.67 kW to 8.911 kW(Fig 12). As shown in Fig 13 there is a diminishing trend in the coefficient of performance (COP) with rising compressor speed suggests a nuanced trade-off between cooling efficiency and energy consumption. Noteworthy is the consistent maintenance of the discharge temperature below 120°C, highlighting the adept temperature control of the compressor. In sum, the data accentuates the imperative of striking a delicate equilibrium in optimizing the performance of a belt-driven compressor, taking into account the intricate interplay among cooling capacity, power consumption, and efficiency across varying operational speeds.

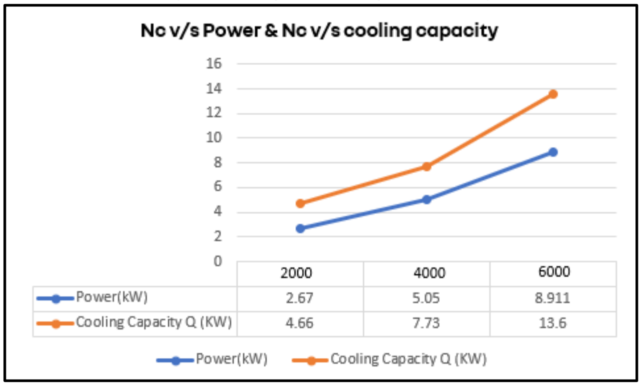


Fig.12 Nc v/s Power(kW) & Nc v/s Cooling capacity for test condition 2

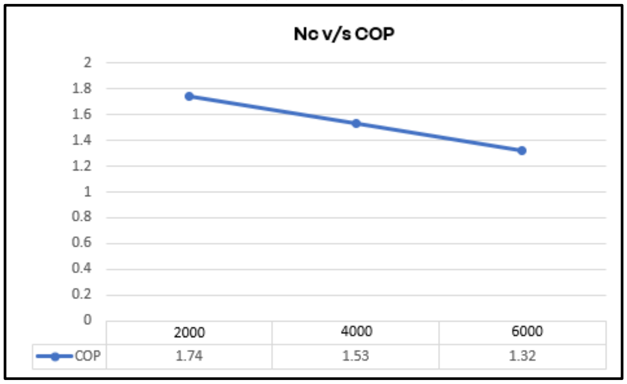


Fig.13 Nc v/s COP for test condition 2

Performance Analysis of e-compressor:

In test condition 1 as shown in Table 7 the Pd/Ps ratio(Pressure Ratio) is taken as 1.5/0.3. The performance evaluation of a belt-driven compressor under specified temperature differentials, denoted as S/H 10K and S/C 5K, with varying compressor speeds are discussed. As shown in Table 7 the rotational speed (Nc) of the compressor increases from 2000 RPM to 6000 RPM, the power consumption also rises from 0.67KW to 2.01KW(Fig.14). This suggests that higher rotational speeds demand more electrical power for the compressor operation.

Table 7. Performance test results of test condition 1 with Pd/Ps = 1.5/0.3

Condition	Pd/Ps (Mpa)	Nc(RPM)	Cooling Capacity Q (KW)	Power L (kW)	COP	Discharge T(°c)
S/H 10K S/C 5K	1.5/0.3	2000	1.484	0.67	2.215	<120
		4000	3.062	1.34	2.285	<120
		6000	4.239	2.01	2.109	<120

The cooling capacity (Q) of the compressor also increases from 1.484kW to 4.239kW (Fig.14) with higher rotational speeds. This is expected, as a faster rotation allows the compressor to remove more heat from the system, resulting in increased cooling capacity.

The Coefficient of Performance (COP) provides insights into the efficiency of the compressor. While there is a slight decrease in COP from 2.215 to 2.109 (Fig.15) as rotational speed increases, the values remain relatively close. This indicates that, despite higher power consumption at higher speeds, the compressor is still effective in producing cooling capacity.

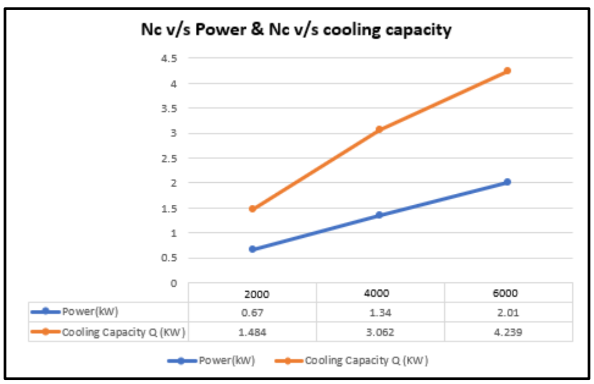


Fig.14 Nc v/s Power(kW) & Nc v/s Cooling capacity for test condition 1

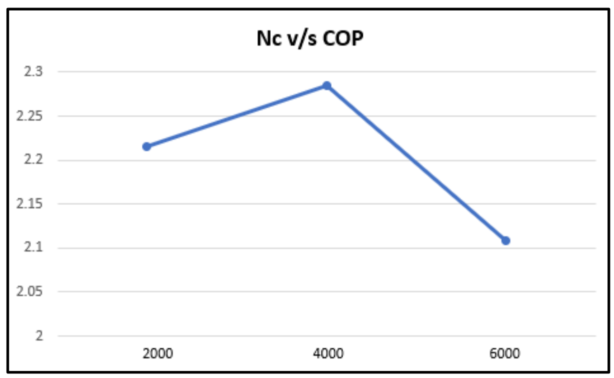


Fig.15 Nc v/s COP for test condition 1

In test condition 2 as shown in table 8 the Pd/Ps ratio(Pressure Ratio) is taken as 2.1/0.4. The performance evaluation of a belt-driven compressor under specified temperature differentials, denoted as S/H 10K and S/C 5K, with varying compressor speeds are discussed.

The pressure ratios of 2.1/0.4 delineate the compression characteristics. As shown in the Table 8, when the rotational speed (Nc) increases from 2000 RPM to 6000 RPM, the power consumption of the compressor also rises 0.916kW to 2.714Kw (Fig.16).

This indicates a positive correlation between rotational speed and power consumption, suggesting that higher speeds demand more electrical power for compressor operation.

Table 8. Performance test results of test condition 2 with Pd/Ps = 2.1/0.4

Condition	Pd/Ps (Mpa)	Nc(RPM)	Cooling Capacity Q (KW)	Power L (kW)	COP	Discharge T(°c)
	2.1/0.4	2000	1.633	0.916	1.783	<120

S/H 10K S/C 5K		4000	3.384	1.84	1.839	<120
		6000	4.67	2.7147	1.699	<120

Like power consumption, the cooling capacity (Q) of the compressor increases from 1.633kW to 4.67kW (Fig.16) with higher rotational speeds. This implies that as the compressor operates at faster speeds, it can remove more heat from the system, resulting in increased cooling capacity.

The Coefficient of Performance (COP) shows a decreasing trend from 1.783 to 1.699 (Fig.17) as the rotational speed increases.

A lower COP at higher speeds indicates that, while the cooling capacity increases, the efficiency of the compressor in converting electrical power to cooling capacity decreases. This might be due to increased internal losses or inefficiencies at higher speeds.

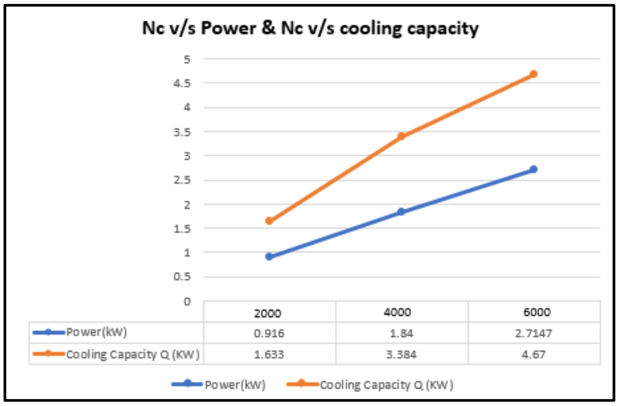


Fig.16 Nc v/s Power(kW) & Nc v/s Cooling capacity for test condition 2

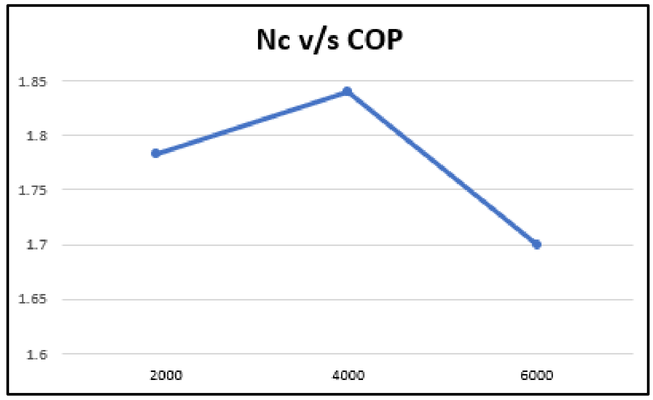


Fig.17 Nc v/s COP for test condition 2

In conclusion, the performance analysis of the e-compressor under various test conditions indicates its responsiveness to changes in speed and pressure ratios. The e-compressor demonstrates a consistent capacity to provide cooling, with corresponding power inputs varying at different operating speeds. Notably, the system exhibits efficiency across the evaluated conditions, showcasing its adaptability to diverse scenarios. The analysis underscores the e-compressor's ability to maintain performance within acceptable temperature limits, highlighting its reliability and suitability for a range of operational settings.

Inferences:

As shown in Fig.18 the evaluation of the Coefficient of Performance (COP) underscores the consistent superiority of the electrically driven compressor. The COP values with compressor speed range of 2000 to 6000 RPM for the electrically driven compressor range from 1.904 to 2.062, outperforming the conventional compressor with compressor speed range of 2000 to 6000 RPM with COP values ranging from 1.313 to 1.632.

This signifies the electrically driven compressor's more effective conversion of electrical input into cooling output, emphasizing its enhanced energy efficiency. The discernible advantage in COP strengthens the case for considering the electrically driven compressor as a preferable option, particularly in applications prioritizing energy efficiency and operational cost-effectiveness.

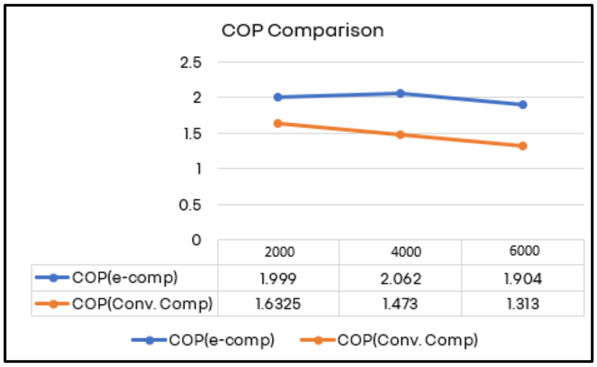


Fig.18 COP comparison between e-compressor and belt driven compressor

As shown in Fig.19 the examination of cooling capacity highlights the divergent performance characteristics of the two compressors. The conventional compressor exhibits a steady escalation in cooling capacity, ranging from 3.68 kW to 10.75 Kw with compressor speed range of 2000 to 6000 RPM.

Meanwhile, the electrically driven compressor with compressor speed range of 2000 to 6000 RPM maintains a more restrained range, from 1.55 kW to 4.45 kW. Despite the smaller cooling capacity range, the electrically driven compressor remains competitive, suggesting a focused and efficient conversion of power to cooling. This nuanced difference in cooling capacity profiles may influence the selection of compressors based on specific cooling requirements and efficiency considerations.

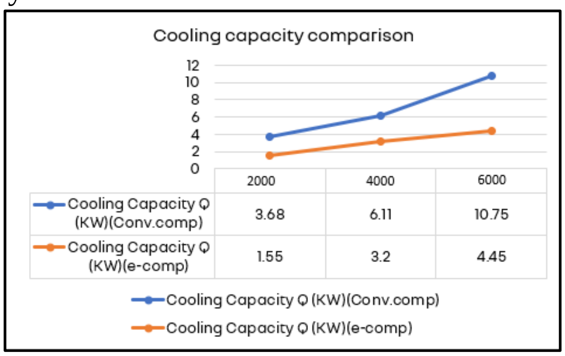


Fig.19 Cooling capacity comparison between e-compressor and belt driven compressor

The analysis of power consumption as shown in Fig.20 reveals a striking contrast between the conventional compressor and the electrically driven compressor. The conventional compressor with compressor speed range of 2000 to 6000 RPM demonstrates a progressive increase in power consumption from 2.22 kW to 7.25 kW across the measured instances.

In contrast, the electrically driven compressor with compressor speed range of 2000 to 6000 RPM consistently maintains lower power consumption levels, ranging from 0.79 kW to 2.36 kW. This discrepancy underscores the inherent efficiency of the electrically driven compressor in converting electrical power to mechanical work, suggesting a notable advantage in energy conservation.

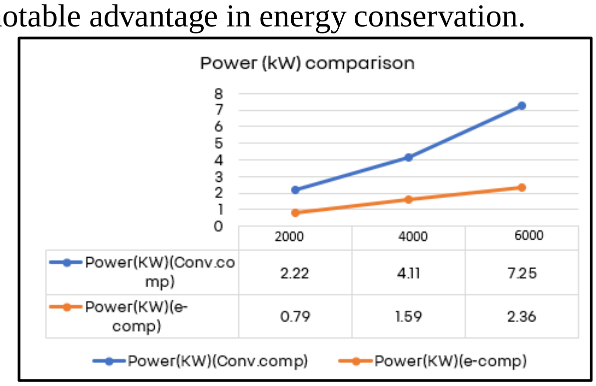


Fig.20 Power comparison between e-compressor and belt driven compressor

Cost Analysis

Comparing the integration of an e-compressor in the entry segment vehicle with the conventional belt-driven compressor, there is a cost increase of approximately 63% when e-compressor is integrated. This marginal increase in the cost is mainly on the part cost . With the integration of e-compressor, other e-components are also integrated for seamless working of the compressor. The other e-components include the DC-DC invertor and the belt-driven starter generator and harness routing for the modifications carried out in all the electrical components. In conclusion, while the upfront cost difference between e-compressors and traditional belt-driven compressors is evident, it is essential to consider the broader spectrum of advantages associated with adopting cutting-edge e-components. Though the Cost up Value is Higher for any Entry Segment Vehicle ,in terms of long-term cost efficiencies to technological advancements, these factors collectively contribute to the overall value proposition of Replacing electric Compressor in entry Segment vehicles.

Conclusion

This study delves into the meticulous selection of compressors, examining their seamless integration into entry-level vehicles. The architecture packaging status indicates a strategic inclusion of the belt-driven compressor within the powertrain unit to optimize assembly feasibility. Conversely, the e-compressor is mounted on the Body in White (BIW) to alleviate potential NVH (Noise, Vibration, and Harshness) concerns that could affect performance. The transition from a belt-driven to an e-compressor necessitates notable modifications, incorporating advanced electrical components like the DC-DC inverter and the belt-driven starter generator. The performance analysis sheds light on the intricate interplay between rotational speed, power consumption, cooling capacity, and overall efficiency in compressor systems. Cost implications reveal an increase in optimization efforts. Tailoring strategies to specific application requirements considering energy efficiency, packaging constraints, and performance targets is crucial. This replacement of conventional compressor by e-compressor can be a potential USP and could further help in contributing to the recent trends in global EV market. This study recommends ongoing research and analysis to address efficiency challenges, particularly at higher rotational speeds, and to enhance the overall performance of compressor systems in diverse automotive applications.

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